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A CAPILLARY ASSISTED THERMOSYPHON FOR SHIPBOARD ELECTRONICS COOLING

by

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ABSTRACT

Recent advances in capillary pumped loop technology were incorporated into the design of a vertical flat plate evaporator for cooling high power electronics aboard naval vessels. This investigation included the design, fabrication, and experimentation of an evaporator plate configured for installation into a standard Navy electronics cabinet. Combining both the characteristics of a Capillary Pumped Loop (CPL) and a thermosyphon, this design integrated the wick from a CPL and the gravity head associated with a thermosyphon. The vertical evaporator plate was designed using AutoCAD, modeled in I-DEAS, and fabricated in a computer-numerically controlled (CNC) milling machine.

The effectiveness of the resulting capillary-assisted thermosyphon (CAT) loop was then evaluated by investigating the effects of subcooling and orientation [tilt and pitch] on the thermal performance. The subcooling effects were studied for a range of heat loads by varying the chill water flowrate between .375 and 3.5 GPM, and varying the chill water temperature between 4 and 37 degrees Celsius. The heat inputs were also varied between 250 and 3200 W. Chill water temperatures were chosen that simulate the conditions of the sea that a warship would be operating in anywhere between the Artic and the Persian Gulf. Subcooling was shown to be dependent upon chill water sink temperature and not strongly dependent upon chill water flowrate. The influence of orientation on the thermal performance of the evaporator plate was also tested by pitching and tilting the plate up to forty-five degrees of inclination and observing the operating temperatures inside the plate. Results showed that the plate operated under all tilt angles, pitch angles, and pressure restrictions imposed. The most significant influence was observed during pitch angle testing due to the geometry of the plate.

KEYWORDS: electronics cooling, capillary pumped loop, thermosyphon, loop heat pipes

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NOMENCLATURE

A_s	Surface Area (m^2)
A	Area (m^2)
A_t	Cross Sectional Area of the Wick (m^2)
CAT	Capillary Assisted Thermosyphon
COTS	Commercial Off the Shelf
CPL	Capillary Pumped Loop
D	Diameter (m)
f	Frictional factor
g	Acceleration due to gravity
h	Heat Transfer Coefficient for Convection (W/cm^2K)
h_{fg}	Latent Heat of Vaporization (kJ/kg)
k	Ratio of Specific Heats
K_{eff}	Permeability of the Wick
K_L	Minor Loss Geometry Factor
L	Length (m)
GPM	Gallons per Minute (Volumetric Flowrate)
m	Mass Flow rate of fluid (kg/s)
q	Heat Rate (W)
P	Pressure (kPa)
R	Ideal Gas Constant
r_{eff}	Effective Pore Radius (μm)
T_s	Temperature of the Surface (K)
T_{sub}	Subcooling Temperature (C)
T_∞	Mean Temperature of Surrounding Fluid
U	Overall Heat Transfer Coefficient for Heat Exchangers (W/m^2K)
UA	Heat Transfer Coefficient including Area (W/K)
V	Velocity (m/s)

Greek Symbols

γ	Specific Weight (kN/m^3)
ρ	Density (kg/ m^3)
ρ_l	Density of the liquid
ρ_v	Density of the vapor
σ	Surface Tension (N/m)
θ	Tilt (Degrees)
θ_c	Contact Angle (rad)
υ	Kinematic Viscosity (m^2/s)
μ	Dynamic Viscosity (Ns/m^2)

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1.0 INTRODUCTION

The future capabilities of naval ships will be directly related to the electronic components used in advanced radar systems, fire control systems, electric propulsion and even electric weapons. Modern electronics continue to grow in speed and functionality but shrink in size and mass, causing the power density to dramatically increase. Thermal management is becoming a major issue for the modern electronic Navy.

Traditionally naval warships were not designed with regard to thermal management outside the engine room and other engineering spaces. The current generation of warships was designed around the traditional standard of having separate turbines for propulsion and for shipboard electricity generation. The propulsion turbines spin a shaft connected to a series of reduction gears, which steps the speed of rotation of the shaft down to a manageable angular velocity suitable for turning the propeller. The concept of using a primary shaft and reduction gears has been the Navy standard since the days of John Ericsson and the *USS Monitor* during the Civil War. The next generation of Naval warships will fall under the concept of an electric ship, where turbines convert all the power produced by the engine into electricity. This electrical power can be distributed as the crew sees fit given its mission and operating profile. The electrical power produced by these turbines will be allocated to the shipboard computers, radars and motors for propulsion instead of having separate turbines to power each individually. The Electric Ship Technology Vision for the United States Navy is:

> Future Warships and combat vehicle machinery systems will provide war fighters the capability to make mission-based tactical allocations of total installed power among weapons, sensors, propulsion, and mission loads. Breakthroughs in power generation, propulsion and distributed systems machinery technologies will provide the critical concepts, architectures, and systems to enable this revolutionary war fighting advantage. (Kuszewski, 2002)

These revolutionary electrical systems will require intensive cooling of their internal electronics in order to function properly. The Navy currently possesses three different ways to dissipate heat from a warship: chilled water, salt water/ fresh water heat exchangers, and using the ship's air conditioning system (Kuszewski, 2002). Chilled water systems are currently the system of choice for cooling high power electronics and can be modified to become more efficient. Salt water/ fresh water heat exchangers are very convenient when the heat load is highly localized, for example the heat removal from a condenser. The simplest and most economical method of removing heat from a limited number of electronics is to use the air inside the compartment. The electronics heat up the air in the room, which then must be cooled using the ship's air conditioning system. However there are two fundamental problems with the use of air conditioning. First, there are limits to the amount of heat that air conditioning can remove from the surface of the electronics. Figure 1 shows a visual representation of the number of shipboard chillers that must be added using current electronics cooling technology to keep pace with the new more powerful electronic systems.

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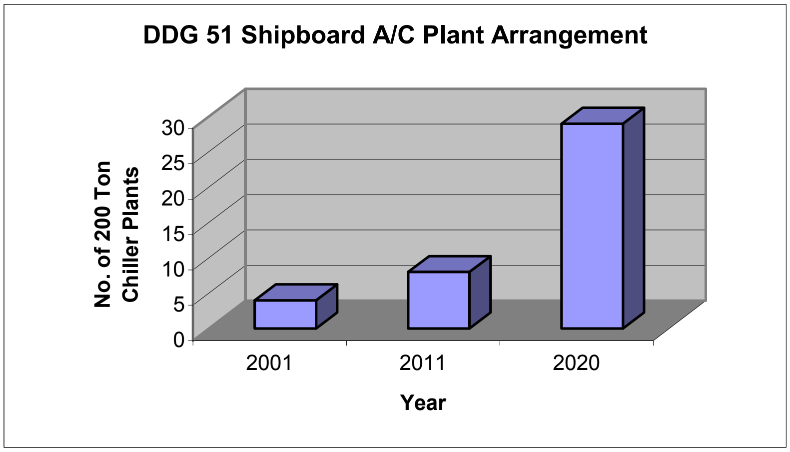
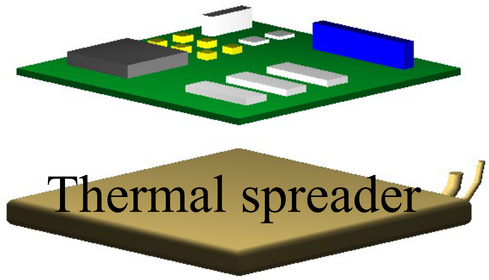


Figure 1: Projected A/C Plant Arrangement of a DDG 51 Class Destroyer (Kuszewski 2002)

There is yet another problem with continuing to use air for cooling electronics. Forced air convection will not remove the heat fast enough. The current Navy high power electronics dissipation requirement established by the Office of Naval Research (ONR) is 3.39 W/cm^2. By 2005 ONR is anticipating localized heat fluxes reaching 1000 W/ cm^2. The difference between the two numbers is on the order of 3 times the current magnitude and cannot be reached by using forced air convection.

The addition of more chillers is not a feasible approach for a warship at sea since there is only a finite amount of space on the ship. Currently there are four air conditioning plants aboard a DDG-51 class destroyer. With the Navy's current electronic cooling philosophy, to cool all of the new technology coming on line, additional chiller plants would have to be added. A more efficient way of cooling electronics must be developed that can fulfill not only the cooling demand that the Navy would place on it, but also be robust enough to withstand life at sea and battle damage. ONR is investigating several different approaches that would solve the thermal management issues facing the modern Navy.

One of the more feasible approaches is the application of capillary pumped loops (CPL) to cool electronics. Capillary Pumped Loop technology will incorporate the same chilled water systems that warships currently have but apply the chilled water in a much more efficient manner. Combined with the new electronics cooling and thermal management initiatives, the Navy has recently developed standardized electronics cabinets. These cabinets, of which there are three different sizes, will be placed aboard every warship in the Navy taking advantage of commercial off the shelf (COTS) technology. An example of next generation electronic cabinets is shown in Figure 2.



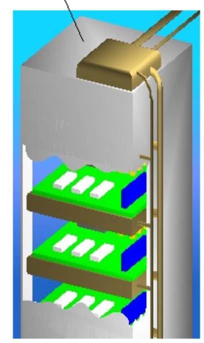


Figure 2: Next Generation Cooling Electronics Cabinet. The thermal spreader will be replaced by a flat plate evaporator using CPL technology. (Kuszweski)

The base of each shelf in the cabinet serves as the heat sink for the electronics which are mounted on the plates. The various plates, called thermal spreaders in the figure are attached in parallel to a single condenser at the top of the cabinet. This figure serves as a conceptual sketch for ONR on the direction they might like to head in electronics cooling.

Currently the standard method of electronics cooling utilizes forced air convection. The concept of cooling electronics by blowing cold air over the hot surface has been around since the advent of electronics and computing. These cabinets have provision for heat exchangers located in the top of the cabinet and fans at the bottom. Air is blown up past the electronics cards into the heat exchanger. The cold air is then returned to the bottom of the cabinet after being cooled in the heat exchanger. The air in the cabinet is sealed from the outside air blowing through the heat exchanger. This means that each cabinet possesses two separate loops that work together to keep the electronics cool. Most of the electronic cards and systems in the cabinet have fins and heat spreaders to increase the surface area available for convection cooling. Figure 3 shows a typical cross section of one of the electronics cabinets.

Since the cold air enters at the bottom and is heated as it rises, there will be a temperature gradient through the air in the electronics cabinet. This temperature gradient within the cabinet is not desirable because the most effective cooling region is located only in the bottom of the cabinet. In addition this system is still limited by the rate at which heat can be removed using forced air convection.

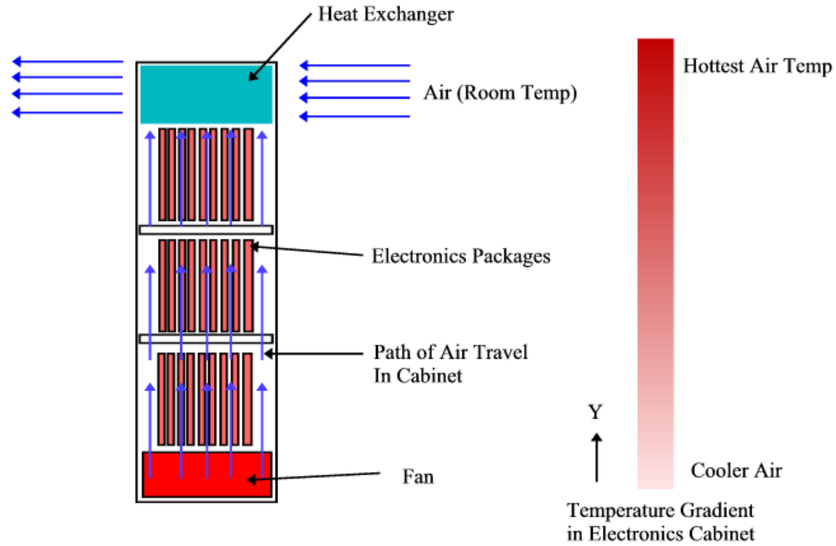


Figure 3: Typical Conventional Electronic Cabinet, cooled using forced air convection.

Much higher heat transfer rates can be achieved by using liquid coolants, and especially by taking advantage of boiling heat transfer. Boiling heat transfer involves using the heat from electronics to evaporate a liquid rather than just raising the temperature of a liquid. A chart showing the magnitudes of different types of heat transfer modes is shown in Figure 4.

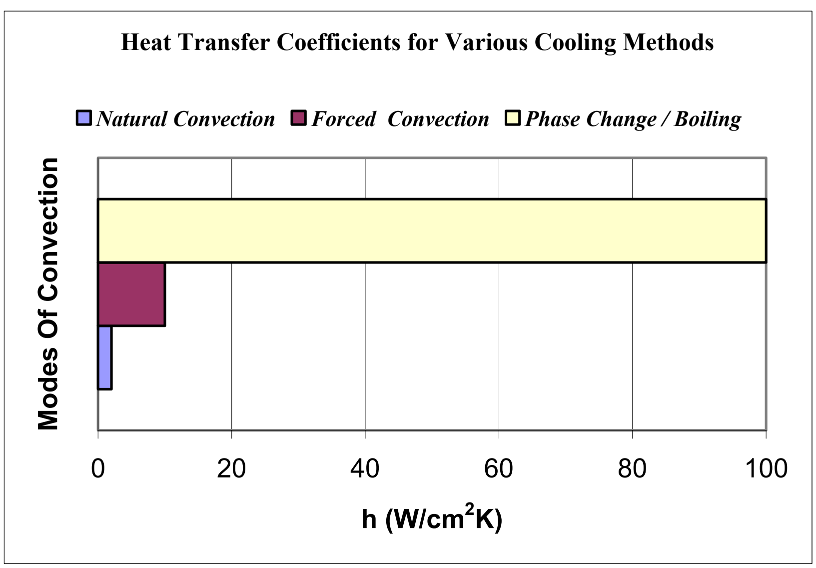


Figure 4: Cooling potentials for various cooling techniques (Kuszewski 2002)

Capillary pumped loops take advantage of the high heat transfer rates that occur during boiling and condensation heat transfer. Figure 5 is a conceptual sketch of a standard electronics cabinet cooled not by convection but by a series of capillary pumped loops together in parallel.

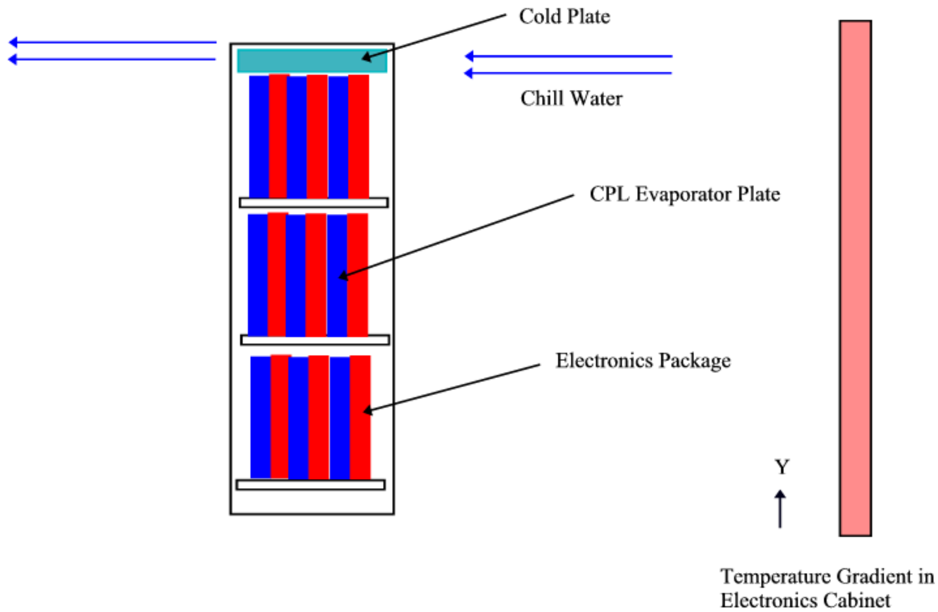


Figure 5: Electronics Cabinet Cooled by Parallel CPL Loops. The CPL system allows the interior of the cabinet to be maintained at a uniform temperature.

With the capillary pumped loop systems integrated into the cabinet, a constant temperature throughout the cabinet can be maintained. This allows electronic packages to be placed in the cabinet in any position or at any height. Thus higher heat flux electronics no longer have to be located near the bottom of the cabinet. The two separate air loops have been removed and replaced by one condenser or cold plate located in the top of each cabinet. A series of evaporator plates, connected in parallel with the cold plate, are used to remove the heat from the electronic components.

This Trident Project designed a new loop heat pipe called a Capillary Assisted Thermosyphon (CAT) loop. The CAT loop combines the capillary pumping head with a gravity head produced by a change in height. The CAT loop allows much higher mass flow rates to be obtained than simple CPL loops. The differences between the two loops and more specifics will be examined later.

The CAT is a closed loop system that keeps the chill water and system water separate to prevent contamination and corrosion from entering the loop or affecting the electronics. This system of loops would be standardized to fit in the COTS cabinets for easy replacement and repair.

The Navy's need for advanced cooling techniques cannot be overstated. Complicated electronic systems like AEGIS depend on the ship's air-conditioning and chill water to stay cool. In event of a chemical, nuclear, or biological attack on the ship, the air conditioning would be shut down

to prevent the intake and spread of the contaminants. The AEGIS technicians aboard the *USS Decatur* stated that they would have fifteen minutes to shut their electronics down before the heat that they produced caused them to melt down. In fact over 55% of all electrical equipment failures can be attributed to heat related causes. (Ohadi, 2002). This means that just over half of all of the electronics that fail all over the world each day, fail because they were inadequately cooled. If the destroyer were still in a combat environment, all of her advanced electronics would become ineffective. Using the CAT system just described, the air conditioning could be secured without any loss in performance because the loop operates solely upon the heat given off by the electronics and chill water systems.

In addition a cooling system based upon the idea of utilizing capillary action as the primary driving force would remove the need for fans and pumps to drive or assist in the cooling operation. One of the major sources of noise aboard warships and submarines are the cooling fans and pumps. Their removal would allow submarines and surface vessels to lower their detection threshold. Unlike the typical way the cabinets are cooled, CAT systems do not draw parasitic power from the ship, nor do they produce additional heat of their own like a fan motor. Utilizing a CAT cooling scheme would not only increase the ship's thermal performance, it would directly contribute to battle readiness and the survivability of the ship. This Trident project studied the feasibility of adapting CAT systems to cool electronics for the Navy.

1.1 OBJECTIVES

This investigation sought to design and use a vertical flat plate evaporator to cool simulated electronics packages. The system will also need to be able to cool hot spots created when these packages are mounted to the evaporator plate's surface. The goals of this investigation included designing a vertical flat plate evaporator to achieve at least 7 W/ cm^2 of heat removal, and the fabrication of the flat plate evaporator and testing its performance in a series of experiments. These experiments have been designed to help prove the feasibility for the use of CAT loop technology on Naval vessels at sea.

The effect of subcooling on the liquid return line, piping, and evaporator plate was examined by varying the flow rate through the cold plate from 3.5 to 0.38 gallons per minute. For optimum CAT loop performance, the temperature of the liquid should remain near the saturation point and not cooled down to a lower temperature. The saturation point is the corresponding pressure and temperature where the addition of any heat does not cause an increase in temperature or pressure but causes the liquid to evaporate or condense. By minimizing the amount of subcooling present in the loop, a higher mass flow rate of the working fluid can be achieved.

The performance of the vertical flat plate evaporator was also tested under a variety of pitch and tilt orientations up to forty-five degrees to show that the pumping head produced in the wick can overcome the pitch and roll of a ship at sea. Finally, the resulting CAT loop was subjected to pressure restrictions in an attempt to find the maximum pressure head produced by the loop. The maximum pumping head produced by the evaporator plate will give engineers information on how far apart the condenser and the evaporator plate can be located.

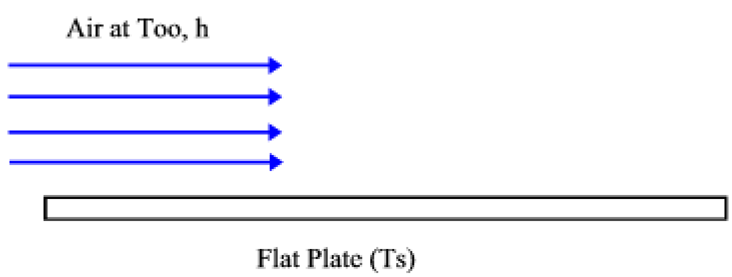
The vertical flat plate evaporator was designed to allow for heat removal from both sides and fit into a standard Navy COTS electronics cabinet. This investigation represented the first attempt by the Navy at designing a CAT evaporator plate for the shipboard use. A successful vertical flat plate design will then be considered for installation into COTS electronics cabinets.

2.0 HISTORY OF ELECTRONICS COOLING

The Capillary Assisted Thermosyphon (CAT) is a derivative of a loop heat pipe, which can be utilized for electronics cooling. The CAT loop is a recent outgrowth of heat pipes and capillary pumped loops. The CAT loop is a new solution to a relatively old problem of removing heat from electronics or other devices that produce unwanted heat during their operation.

For many years the primary mode of cooling electronics has been forced air convection. This cooling mode simply uses air blown by a fan to circulate over the heat source. As electronic packages and circuits were initially large and low in power produced, the large surface area of the packages made cooling by convection a good way to remove heat from the circuit. However, as electronics continue to shrink in size and gain in power, higher heat fluxes result that cannot be removed solely by forced convection. Many current devices use heat spreaders and fins to distribute heat to a larger surface area where the heat can be taken away to ensure adequate cooling. However, these extra devices take up valuable space and weight especially for naval and space applications. Right now, electronics packages are pushing the limits of forced air convection, and larger thermal management problems loom ahead. The evolution of electronics cooling will be briefly discussed to provide perspective on the project.

One of the simplest ways to cool electronics has been the use of forced convection. This method of heat transfer depends primarily upon the fluid used for convection and upon the velocity of the fluid. The two common fluids used for forced convection purposes are air and chilled water. Air convection involves blowing air across the surface to be cooled. If chilled water is utilized instead of air, the chill water is piped next to the heat source and the heat travels through the pipe walls by conduction and into the chill water. The amount of heat transfer, q, depends upon h, the heat transfer coefficient, A_s, the surface area, and ΔT, the difference in the temperatures between the hot surface and the fluid. This relationship can be expressed mathematically as $q=hA_s(T_s - T_\infty)$. Figure 6 is an example of forced air convection from a flat plate.



To increase the heat transferred by forced air convection, heat spreaders and fins have been developed to transfer the heat produced from one small electronics chip to a larger surface area to ensure that the heat is removed from the chip. Forced air convection is approaching its limits of effectiveness as employed for electronics cooling, as heat fluxes rise above 3 W/cm^2 (Kuszewski, 2002). Thus the only variable in the above equation has become the surface area. To increase the amount of heat transferred, the surface area must increase. Aboard a ship space

is at a premium, and thus valuable space and weight is going to be taken up by these heat spreaders and fins.

Using water as the working fluid is more efficient than using air. The convection heat transfer is larger for water than air due to the larger heat capacity of water. Water is cooled to a low temperature in chillers and then passed through pipes that are in thermal contact with the electronics. The heat from the electronics passes through the pipe walls and into the water flowing past. The heat transfer coefficient depends upon the velocity of the water and its temperature. Like the air, the water is only being heated up sensibly by all heat that it absorbs. Unlike air, water cannot directly touch the surface of the electronics because it is an electrical conductor and would short out the electronics being cooled. While more effective than forced air cooling, chill water is limited by the same factors. There is only a small immediate benefit to transitioning to a chill water system that directly cools electronics. To take advantage of the benefits of using chill water, an intermediary system can be engineered that uses another method to remove heat from the electronics and then transfer it to the chill water for removal to the sea. This intermediary system is the CAT loop.

A more effective way of transferring heat involves changing the phase of the working fluid rather than just raising its temperature. Using water as an example, only 4.18 kJ of energy are removed by raising one kilogram of water one degree. By vaporizing the same amount of water, 2257 kJ are absorbed (Cengel, 2002). Thus there is a large difference between simple convection and a cooling mechanism or process involving a phase change. The easiest way to cool electronics by a fluid undergoing a phase change is to place the electronics in thermal contact with the fluid in a closed system. This prevents the fluid from interfering with the operation of the electronics and yet provides the ability to transfer higher heat fluxes. For two-phase systems, the amount of heat that can be removed is related to the mass flow rate of the fluid and its latent heat of vaporization. Figure 7 shows an example of a two-phase electronics-cooling device.

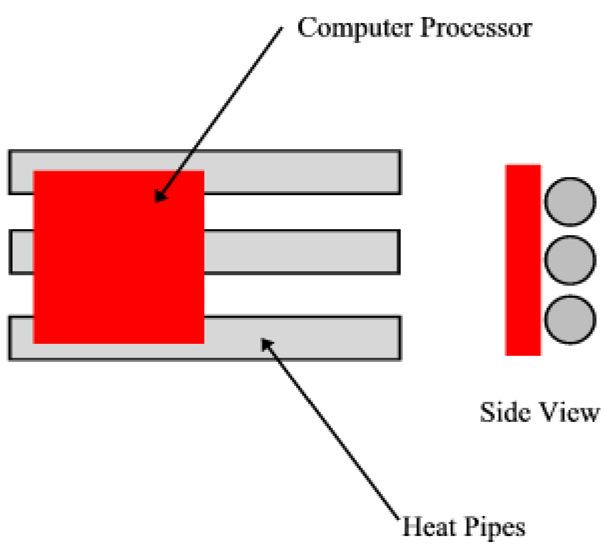


Figure 7: Two-Phase Electronics Cooling

The computer processor is mounted on a device called a heat pipe. In the heat pipe, the working fluid undergoes a phase change as is boiled off underneath the processor and condenses at the other end of the heat pipe where the thermal sink exists. The most common fluid in heat pipes is water because of its stability and temperature range for a phase change. The energy transferred during a phase change process is given by the equation, $\dot{Q}=\dot{m}h_{fg}$, where $\dot{m}$ is the mass flow rate of the fluid and h_{fg} is the latent heat of vaporization for the fluid. Heat pipes are the simplest form of a two-phase approach to electronics cooling. The principles behind their operation will be examined to provide a foundation to discuss loop heat pipes and then capillary pumped loops before moving on to discuss the CAT loop.

2.1 HEAT PIPES

The principles behind heat pipes are related to the concepts of boiling and condensation. Boiling was one of the earliest methods of cooling. Jacob Perkins invented heat pipes in 1836, in which heat transfer was created by a change in phase of a liquid inside of a closed tube (Fahgri 1995). Heat pipes were the first two-phase devices used to transfer heat in a controlled system. The basic design is tubular in form, with one end of the pipe acting as the condenser section and the opposite end acting as the evaporator section. A porous wick connects the two sections and provides the pumping head to ensure the heat pipe's operation. Heat pipes are hermetically sealed, meaning that there is no contact of the outside environment with the fluid present in the heat pipe. The liquid evaporation in the wick in the evaporator zone creates a capillary pumping head which pushes the vapor towards the condenser end. The vapor then condenses in the cooler region and then the wick pulls the condensed liquid back into the evaporator section where the process is repeated. It is important to understand how heat pipes work and their governing theory because CPLs are a modern derivative of them. Figure 8 shows a cutaway view of a heat pipe that shows all of the major components present.

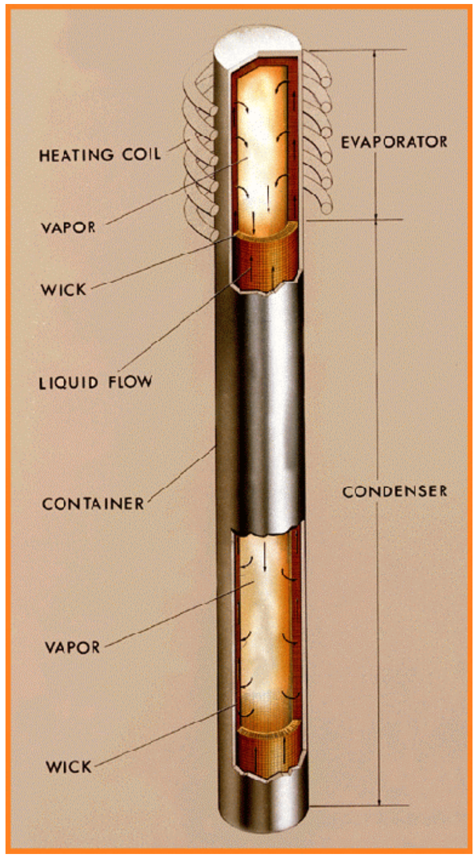


Figure 8: Cutaway View of a Heat Pipe

2.1.1 HEAT PIPE THEORY

The wick in the above figure is the most important part of the heat pipe. The wick creates the change in pressure that causes the fluid to flow from one region to another. This change in pressure is known as the capillary pumping head. The capillary pumping head is directly dependent upon the wick. It is related by the equation below where σ is the surface tension of the working fluid, θ_c is the contact angle that the liquid forms with the pores, and r_{eff}, which is the effective pore radius of the wick itself (Faghri, 1995):

$$\Delta P_{CAP} = \frac{2\sigma\cos(\theta_c)}{r_{eff}} \qquad \text{Eq(1)}$$

The two factors with the largest effect upon the capillary pumping head are the contact angle and the pore radius. As the contact angle increases, the pressure head decreases. The contact angle is a function of the fluid-vapor-solid wetability and cannot be affected by the designer. the contact angle is not affected by the orientation of the wick or the heat pipe. The pore radius for wicks is typically between 10 and 30 μm. The smaller the pore radius the higher the developed pressure head; however, as the pore radius decreases the pressure drop associated with return fluid flow through the wick increases because the fluid must force itself through smaller and

smaller pores. The higher the pressure drop the lower the mass flow rate is through the system and thus less heat is removed. This brings up the relationship between the pressure drops.

The performance of any fluid system is related to the sum of the pressure drops in it. If the pressure throughout the system is the same, no fluid will flow. The basic equation relating the pressures is given below. In a working heat pipe, the capillary pressure defined above is equal to the sum of the pressures around the system including the ΔP_{liq}, the pressure of the liquid in the return lines, ΔP_{vap}, the pressure of the vapor flowing to the condenser section and finally the $\Delta P_{flow\ losses}$ which is the sum of all of the pressure drops caused by the flow through the wick, friction, and rough surfaces in the path of the flow.

$$\Delta P_{CAP} = \Delta P_{vap} + \Delta P_{liq} + \Delta P_{flow\ losses} \qquad \text{Eq.(2)}$$

Now that the equation governing flow through the heat pipe has been developed, the flow must be related to the amount of heat added to the system. The basic equation for fluid flow, Equation 3, was introduced during the introduction to two-phase flow and is listed again below.

$$\dot{Q}=\dot{m}h_{fg} \qquad \text{Eq.(3)}$$

The amount of heat input into the system is related to the latent heat of vaporization and the mass flow rate. In turn, the mass flow rate is composed of several factors including A, the cross sectional area for the flow to travel through, ρ, the density of the fluid, and V, the velocity of the fluid. The resulting equation is thus:

$$\dot{m} = \rho A V \qquad \text{Eq.(4)}$$

Now that the basic equations have been examined, they can be used to explain the performance of the heat pipes. A cross sectional cutaway of a heat pipe is shown below in Figure 9 with four state points. The four state points occur as the fluid starts as a liquid in the evaporator section and follow the fluid as it first is heated and then vaporized. The fluid then travels as a vapor to the condenser section of the heat pipe where it is condensed and the process is repeated. State point one occurs just before the working fluid is pulled up through the wick. The working fluid is a liquid at this stage. After the liquid is pulled through the capillary wick, it begins to be boiled off by the heat applied to the system. This occurs in the evaporator and is listed as state point one. Upon leaving the evaporator as a vapor the working fluid travels to the condenser section. State point two occurs just as the vapor leaves the evaporator. State point three occurs in the condenser section of the heat pipe just before the working fluid condenses to form a liquid again. Finally state point four occurs just after condensation and before the liquid is pumped back to the evaporator section. All of the heat has been removed from the working fluid at this point.

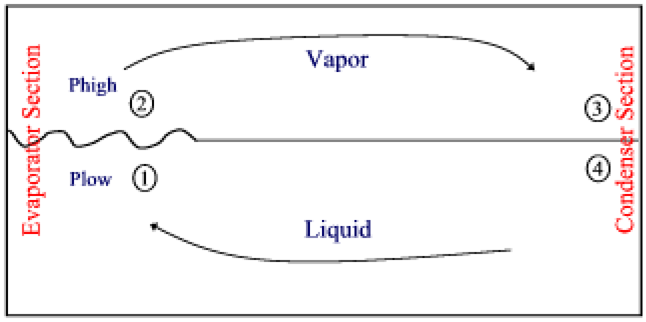


Figure 9: Pictorial View of the Pressure Changes in a Heat Pipe.

The pictorial view shows how the pressures in the evaporator section and condenser section are related and cause the working fluid to flow. The pictorial view in Figure 9 shows the fluid flowing in a clockwise pattern.

A temperature vs. pressure curve for heat pipes is shown below in Figure 10. The curve also shows the saturation line for the working fluid. This curve is generic and shows how pressure and temperature change in a heat pipe qualitatively. The four state points from the pictorial view are also labeled on the curve.

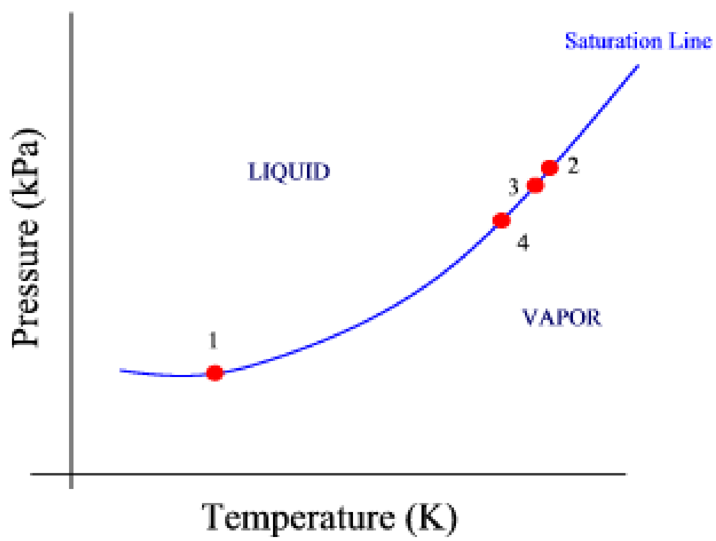


Figure 10: Saturation Curve Produced by Heat Pipe Devices. The blue line represents the saturation curve for the working fluid. The pressure change in the heat pipes is caused by capillary action in the wick.

The saturation curve shows the phase changes that take place as well as the driving force behind the wicks performance, capillary action. Steps 1 to 2 occur in the evaporator section, where vaporization takes place as heat is added to the system. The pressure drop caused by fluid flow pressure drops caused by the wick shows how the pressure drops as the fluid travels from the

evaporator to the condenser region. Finally the fluid is condensed and pumped back by the wick to the evaporator section. The loss of pressure is caused again by pressure drops in the wick created as the working fluid is forced through the small pores.

2.2.1 HEAT PIPE LIMITATIONS

Heat pipes are passive systems, driven by an applied heat flux and capillary action in the wick (Ku, 1997). This means that unlike fans and air conditioners, heat pipes do not draw parasitic power or produce noise, light, or create additional heat. The benefits of these advantages have already been discussed. However, these same benefits also lead to limitations that can dramatically affect the performance of the heat pipe. There are several basic limitations that will be discussed before several specific limitations will be examined.

While the latent heat of vaporization of the working fluid provides an advantage over forced air-convection cooling, heat pipes are limited by their size. The larger the pipe, the larger the wick, which results in a higher pressure drop which in turn causes a lower mass flow rate (Faghri, 1995). The lower mass flow rate means that the heat pipe is not cooling as effectively as it should. The equation for calculating the pressure drop in the wick is given below, where the properties of viscosity, μ, and density, ρ, come from the working fluid. Q_{in} is the applied heat input and L_{eff} is the length of the wick in the heat pipe and A_t is the cross sectional area of the wick This equation is also known as the capillary limit (Faghri 1995).

$$\Delta P_{wick} = \frac{\mu_{liq} Q_{in} L_{eff}}{\rho A_t} \qquad \text{Eq.(5)}$$

It is apparent from the equation that the length of the wick directly increases the pressure drop across it. It is also important to note that as the heat input increases so does the pressure drop because the velocity of the working fluid increases. The pressure drop equation illustrates the fact that the heat pipe is an effective cooling device only for a certain limited range of heat inputs and overall sizes.

Heat pipes are also limited by the fact that the flow is bi-directional; the liquid is being returned to the evaporator section while the vapor is traveling to the condenser section. This means that the liquid and the vapor are flowing against one another. An equivalent experience would be to run continuously into the wind. This bi-directional flow cannot be solved in a conventional heat pipe but can be avoided entirely by using a loop heat pipe.

Another limitation resides in the fact that the evaporator and condenser sections are not thermally isolated. The metal pipe walls with a high thermal conductivity connect them, allowing heat to travel from one region to another by conduction. This makes each section much less effective because of the associated heat transfer between the evaporator and condenser. Because heat pipes are placed in thermal contact with a heat source the walls cannot be made out of ceramics or other insulators. To transfer the heat effectively the walls of the heat pipe must have a high thermal conductivity. Unfortunately the high conductivity also minimizes the difference

between the evaporator and condenser section. This fact brings out several specific heat pipe limitations that will now be examined.

2.2.2 CONDENSER LIMIT

Heat pipes are subject to a condenser limit, where the condenser section, because of its size and not being thermally isolated, can only remove a certain amount of heat (Faghri 1995). The condenser section cannot remove any more heat. This increases temperature throughout the system and can lead to dry out of the wick by boiling off all of the working fluid. The heat transfer rate of the condenser is governed by the equation below (Incropera, Dewitt, 2002):

$$Q_{out} = UA\Delta T \qquad \text{Eq.(6)}$$

The heat removed is Q_{out}, and it depends upon U, the overall heat transfer coefficient for the condenser, A, the surface area present inside the condenser for heat exchange, and the log mean temperature difference in question.

The electrical device that the heat pipe is keeping cool determines the heat input, Qin, to the heat pipe. As long as it is below or equal to the max Q_{out} produced by the condenser the heat pipe will work. In the condenser heat transfer equation the area and the heat transfer coefficient are fixed quantities and as the temperature difference decreases, less heat can be removed. Once the heat input exceeds the total amount of heat that can be removed the heat pipe will cease to work.

2.2.3 BOILING LIMIT

The boiling limit deals primarily with the wick in the evaporator section. A heat pipe's boiling limit is reached when Q_{in} reaches the point where it begins to boil off fluid faster than the capillary pumping head created by the wick can replace it. Once this limit is reached, it will cause the wick to dry out, ceasing the capillary pumping head, which prevents the heat pipe from functioning. The boiling limit is represented by the following equation (Faghri, 1995):

$$Q_b = \frac{2\pi L_e K_{eff} \Delta T_{crit}}{\ln(R_i/r_{eff})} \qquad \text{Eq.(7)}$$

The Q_b term represents the input heat at which the boiling limit would be reached. The length of the wick, L_e, and the permeability of the wick, K_{eff}, are both set physical properties. The natural log term comes from the ratio of the inner wall radius of the heat pipe and the pore radius. Every heat pipe has a boiling limit and thus cannot be used for heat inputs approaching the boiling and dry out region.

2.2.4 SONIC LIMIT

Another major limitation that heat pipes face is the sonic limit. When the sonic limit is reached, the velocity of the fluid traveling in the heat pipe has reached the speed of sound. The speed of

sound for the working fluid depends upon the temperature of the fluid in question. The equation for calculating the speed of sound for a specific fluid is listed below where k is a fluid property, R is the ideal gas constant, and T is the current fluid temperature (Cengel, 2002):

$$C = (kRT)^{.5} \qquad \text{Eq.(8)}$$

Equation 3, which governs the mass flow rate of a two-phase system, relates Q_{in} to the velocity and raising the heat input causes the velocity of the fluid to increase until it reaches sonic flow , and becomes choked. Heat pipes are not built to withstand the forces and pressures involved with sonic and supersonic flow. Figure 11 shows how increasing the heat flux will cause the working fluid to reach a supersonic velocity.

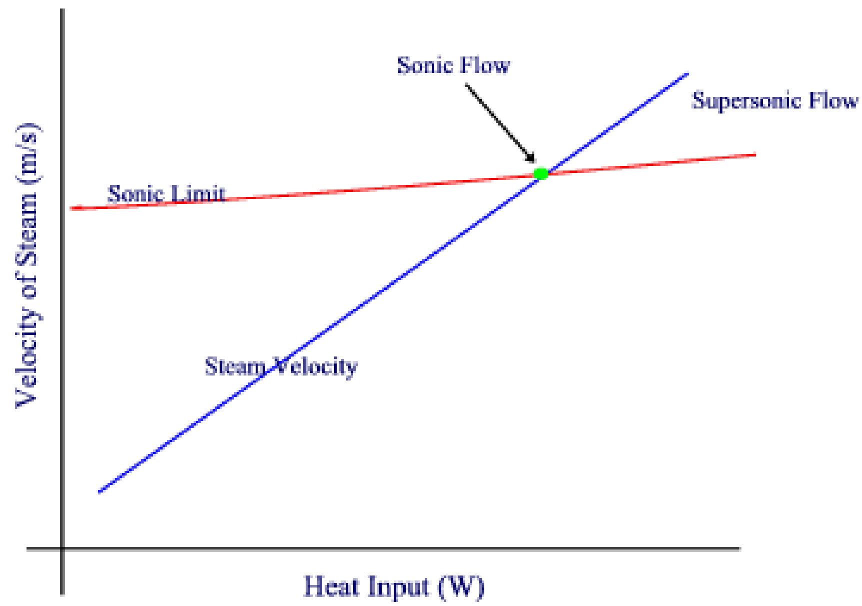


Figure 11: The Effect of Input on Working Fluid and Sonic Limit. This effect is shown qualitatively for water as the working fluid in the Heat Pipe.

Both the steam velocity and the sonic limit increase with the heat input. The heat input controls the velocity of the water in the system and the velocity of the water rises much faster than the sonic limit. Eventually at the sonic limit the speed of the steam in the system equals the speed of sound and sonic flow results, this is shown by the green dot in Figure 11. If more heat is applied the flow velocity continues until the flow becomes choked and the choked mass flowrate for a given h_{fg} will ensure that higher heat input levels cannot be met and the heat pipe will begin to overheat and dry out.

2.2.5 HEAT PIPE CONCLUSIONS

Despite these limitations heat pipes are very effective devices for transferring heat and cooling electronics. They are especially effective in providing cooling for computer processors and other small high heat flux devices that cannot be cooled by forced convection alone. While heat pipes are subject to numerous limitations, their ability to harness a controlled phase change increases their effectiveness and allows high heat flux devices to be constructed and operated.

2.3 THERMOSYPHONS

Another popular method of cooling electronics using a controlled phase change is a thermosyphon. A thermosyphon is a vertically oriented heat pipe without the wick but with a liquid pool at the bottom in the evaporator section (Faghri, 1995). Thermosyphons have been popular choices for thermal management because, like heat pipes they are totally passive devices but instead of relying upon the capillary pumping head produced by a wick, they rely upon a gravity head to ensure operation. Without the wick present, the thermosyphon is also subject to a different set of limitations. A cross sectional view of a thermosyphon is in Figure 12.

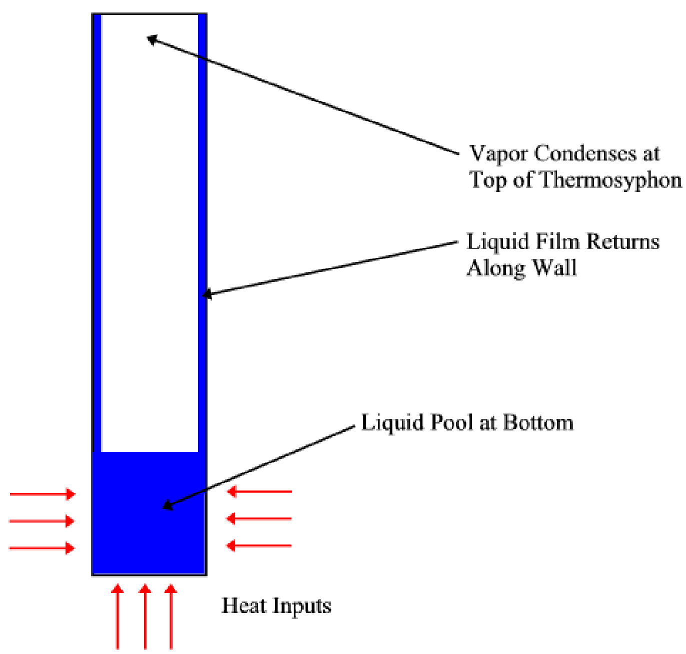


Figure 12: Thermosyphon Cross Section

Thermosyphons have a simpler design and a simpler theory governing their operation than heat pipes, because only the gravity head drives the flow.

2.3.1 THERMOSYPHON THEORY

The total heat transfer coefficient in thermosyphons is determined by the falling liquid film and the counter current vapor flow (Faghri, 1995). Equation 3, $Q_{in} = mh_{fg}$, again governs the flow in a thermosyphon. However, the change in pressure that allows its operation is defined by the following equation where γ is the specific weight of the working fluid and Δh represents the change in height between the evaporator section and the condenser section:

$$\Delta P_{grav} = \gamma_{liq}(h_{cond} - h_{evap})\cos(\theta) \qquad \text{Eq.(9)}$$

For a thermosyphon to work properly the condenser section must be located above the evaporator section because it creates the gravity head present in the system. If the thermosyphon is tilted or pitched, the gravity head is reduced accordingly. Only this gravity head drives the system and thus their use at sea is questionable because a ship is never in a perfect vertical orientation. The cosine term in the above equation shows how the pitching lowers the gravity head. A thermosyphon cannot be used in an aeronautical or a space application because of its dependence upon gravity.

2.3.1 THERMOSYPHON LIMITATIONS

While practical devices for cooling, thermosyphons also have several different limitations. They cannot be used in as wide of range of applications as heat pipes because they lack the pumping head created by the wick. Therefore thermosyphons are limited not only by their gravity head but also by fluid dynamics. One example of this is the transition from laminar to turbulent flow. As heat inputs increase in intensity, the flow along the walls of the thermosyphon tube becomes turbulent, resulting in a higher shear stress. If the shear stress continues to increase the fluid will stop flowing and the thermosyphon will dry out. The max heat input into a thermosyphon is governed by the following experimentally derived equation:

$$q_{max} = .142\rho_v^{.5}(g\sigma(\rho_{liq}-\rho_{vap}))^{.25} \qquad \text{Eq.(10)}$$

The max heat input is related to the density, ρ, of the liquid and vapor, along with the acceleration due to gravity and the surface tension of the fluid. This max heat input is independent of the thermosyphon geometry. After this heat input is reached the thermosyphon will begin to boil off the working fluid faster than it can be replaced by dripping down the pipe walls.

With the evaporator and condenser still in the same body, all of the condenser limitations from heat pipes also still apply to the thermosyphon because of the limited region for condensation and the fact that for effective heat transfer the walls of the thermosyphon must have a high heat transfer coefficient.

The flow of the vapor and the liquid are still in opposition to one another and thus increase the fluid flow losses in the system and decrease the performance of the thermosyphon. The sonic

limit is also another limitation that a thermosyphon, like all two-phase heat transfer devices, faces.

2.3.2 THERMOSYPHON CONCLUSIONS

A thermosyphon is another effective way to cool electronics. It is in many ways a simpler version of the heat pipe. While it does not possess an innate pumping head, it also is not subject to as many pressure drops and system losses and can thus achieve a higher mass flow rate than a heat pipe. Thermosyphons are also not subject to as many dimensional constraints as well, the boiling limit for a thermosyphon being completely independent of the thermosyphon geometry. Like heat pipes, utilizing thermosyphons is an innovative way of cooling electronics. However, thermosyphons have limited application to the Navy because they operate solely by a gravity head.

2.4 SPRAY COOLING

The U.S. Navy has been actively looking at several revolutionary ways to cool electronics. One of these revolutionary ways has been the advent of spray cooling. Spray cooling operates upon the premise of simple boiling. A dielectric, (nonconducting) fluid is to be sprayed upon the electronics in an open system. The dielectric fluid coats the electronics in a thin film where it is then boiled off by the heat generated by the electrical devices and recycled again at the top of the electronics cabinet. Spray cooling is almost five hundred times more effective than simple convection (Kuszewski, 2002). The theory behind this approach is the simplest to understand because it is just thin film boiling. Spray cooling is equivalent to human sweating to keep their body temperature under control. The issues with spray cooling lie not with its effectiveness but over its implementation into an actual working device. Aside from forced air convection, the previous two-phase heat transfer devices are all closed system designs. The working fluid is kept completely separate from the electrical devices that it is to keep cool. Spray cooling proposes to coat the electronics with a thin film that is in direct contact with the electronics. An example of spray cooling is illustrated in Figure 13.

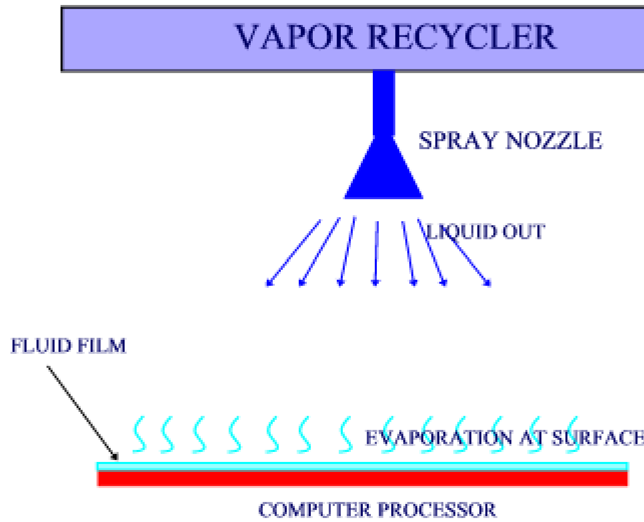


Figure 13: Spray Cooling Illustration

Spray cooling is excellent for localized high heat flux cooling because the thin film of liquid coating the surface of the electronic device is held at a constant temperature because of the evaporation. The pressure inside the cabinet will also be maintained at a constant atmospheric pressure.

With its advantages and simplicity, spray cooling also suffers from several limitations. Spray cooling works best on high heat fluxes; it doesn't cool as effectively for low heat flux devices because the temperature of the devices is not enough to ensure a constant rate of evaporation. This would lead to the film of liquid growing into a puddle and convection cooling rather than a phase change. The actual mechanics and logistics are also a major disadvantage. Most liquids are excellent conductors of electricity and thus the electronics in question need to be altered or designed to withstand a liquid unless dielectric liquids are used. This means that every electronic device would need to be redesigned or altered so that the application of a fluid would not cause the electrical circuit to short out unless specially adapted for a dielectric. Several initiatives are currently being sponsored by the Navy in studying the application of spray cooling devices.

2.5 LOOP HEAT PIPES

Because of the limitations already described, the natural evolution of the heat pipe and the thermosyphon is the loop heat pipe (LHP). A loop heat pipe has a separate evaporator and condenser that are connected by tubing. By separating the two regions, limitations that faced thermosyphons and heat pipes can be minimized. For heat pipes, the geometry of the condenser and evaporator is important in determining the heat pipe's limits. In a LHP, the evaporator and condenser can be independently tailored to suit the application. A typical loop heat pipe is shown in Figure 14.

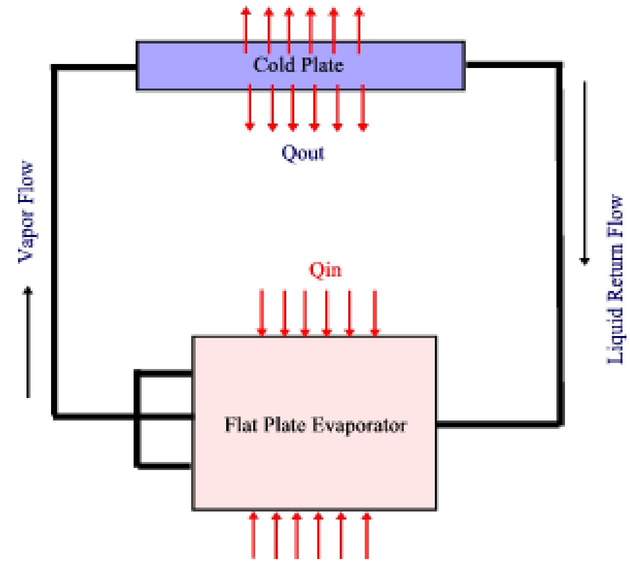


Figure 14: Loop Heat Pipe Schematic. This particular loop has a cold plate for the condenser and a flat plate evaporator.

The first loop pipe was successfully created in the Urals Technical University in the Soviet Union in 1971 (Faghri, 1995). Loop heat pipes are much more complex than the original heat pipes. The evaporator and condenser have now been separated and thermally isolated. The wick, which once traveled the length of the heat pipe, is restricted to the evaporator section only. By limiting the wick to the evaporator section, a much higher mass flow rate could be produced, which means that the loops could be applied to cool larger, more powerful devices that produce a higher heat flux. The evaporator and condenser are now connected by liquid return lines and vapor transport lines. Most loop heat pipes fall under the more specific category of capillary pumped loops, CPLs, because the capillary pumping head generated by the wick is the primary driving force throughout the wick.

One of the main forces behind the creation of LHPs was a direct result of their application to space. The capillary action inherent in the wicks associated with the pipes meant that the loss of gravity would not affect their pumping action, unlike typical thermosyphons. Because loop heat pipes are larger than heat pipes, more heat can also be effectively transported away from the delicate and complex electronics packages. Loop heat pipes and spray cooling are the two primary initiatives being looked into by the Navy for future cooling needs.

2.5.1 LOOP HEAT PIPE ADVANTAGES

The wick is limited only to the evaporator section and thus it limits the pressure drop through the evaporator. A smaller pressure drop means a higher mass flow rate throughout the system. This in turn means that more heat can be removed from the electronics in question. Instead of flow lengthwise through the wick, for LHPs, the fluid usually flows through wicks less than an eighth of an inch thick and the flow is unidirectional, meaning that the liquid and the vapor travel in the same direction in a loop. Unidirectional flow allows for less resistance to flow of the working fluid as well as decreasing the amount of turbulence and shear stress present in the loop. Heat is

still added in the evaporator and removed by the condenser but each can be separately tailored for the specific task. While the condenser, capillary, sonic, and boiling limits are still design concerns; they can be addressed individually rather than facing tradeoffs present in choosing a design for a simple heat pipe.

Many LHPs also combine characteristics from thermosyphons and heat pipes by placing the evaporator and the condenser at different heights. This allows a gravity pumping head to be created to supplement the capillary pumping head. These pressure heads allow a higher mass flow rate that means that more heat can be removed from the electronics.

2.5.2 LOOP HEAT PIPE LIMITATIONS

LHPs are still subject to all of the same limitations as simple heat pipes. The boiling limit, sonic limit, and capillary limit are all major design concerns, and LHPs cannot be placed in a cooling scenario where one of these limits will be exceeded. Another major limit that was not faced by heat pipes and thermosyphons was the pressure drop through pipe flow. If the loop is too large, meaning that the evaporator and the condenser are located too far away, the pressure drop created by fluid flowing through pipes will cause the loop to stop working. The pressure drop through a pipe is described by the following equation, where f, L, d, K_L are the frictional factor, the pipe length, pipe diameter, and specific pipe geometry respectively:

$$\Delta P_{PIPE\ LOSS} = \frac{fL\rho V^2}{2d} + \frac{K_L V^2}{2g} \qquad \text{Eq.(11)}$$

The length of the pipe is a major factor in determining the pressure drop. The minor drops in pipe flow can all be accounted for by the K_L factor which takes into account all of the bends, valves, and joints of the piping itself. This limitation does not face thermosyphons or heat pipes and thus must be accounted for in the design of the system. This is a significant contributing factor to system losses that must be studied in order for shipboard designers and architects to create LHP systems if they are implemented.

2.5.3 LOOP HEAT PIPE CONCLUSIONS

Loop heat pipes are an improved derivative of heat pipes and thermosyphons. They combine the best features of both in order to produce a controlled two-phase heat transfer device. Loop heat pipes are suitable for a wide range of cooling applications because of their robustness. They face all of the same limits that heat pipes face, but they have the ability to reduce and minimize the limitations based upon the particular design of the evaporator and the condenser. LHPs are proven , consistent devices. Their application to the naval environment must be studied in order to deduce their feasibility for use at sea, cooling electronics and other devices for the U.S. Navy. Loop heat pipes can be incorporated to cool existing electronic devices along with being engineered for future heat loads. The flexibility of the loop heat pipes means that they can be designed to fit in many different spaces under many different conditions.

3.0 CAPILLARY PUMPED LOOP THEORY

The driving force behind loop heat pipes is the capillary pumping action produced in the wick material. The wick material is used to draw the fluid to the evaporation surface and eliminate the need for an external pump. Therefore, these systems are referred to as Capillary Pumped Loops (CPL). The basic structure of a CPL flat plate evaporator is illustrated in Figure 15. Heat is applied at the surface of the plates where it is transferred to the surface of the wick by conduction. The fluid in the wick absorbs a certain amount of heat during the evaporation process. The vapor produced leaves the plate via a vapor channel that allows the vapor to flow towards the condenser. The capillary pumping head produced by the wick plus any additional gravitational head ensures that the evaporated fluid is continually being replaced by liquid in the wick.

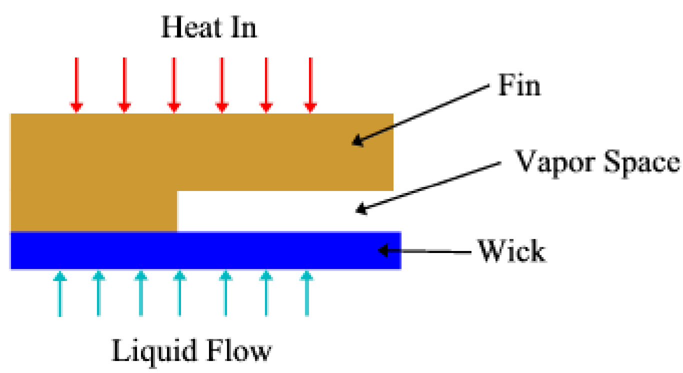


Figure 15: Basic CPL structure (Faghri, 1995)

The word capillary itself has two meanings, small bore or diameter and the force created by surface tension in a fluid. Capillary pumped loops actually use both aspects of the definition. The wick contains thousands of microscopic pores, often on the scale of micrometers or smaller. The working fluid in liquid form is pulled through the wick by the surface tension forces concentrated inside the pores. The contact angle of the meniscus and the liquid in the pores also has an effect on the capillary pumping pressure produced by the wick. The basic equation showing how the capillary pumping head is generated comes from the same derivation as the equation for the capillary pumping head produced by heat pipes (Faghri, 1995):

$$\Delta P_{CAP} = \frac{2\sigma\cos(\theta)}{R_{eff}} \qquad \text{Eq. (12)}$$

where P_{CAP} is the capillary pumping head, σ is the surface tension and R_{eff} is the effective pore radius. This capillary pumping head will be examined in detail to see how the different parameters affect the pressure head produced. After this equation is examined, the other equations and parameters in the theoretical CPL model will be explained.

3.0.1 SURFACE TENSION

The first important parameter in Equation 12 is σ, the surface tension of the working fluid. Surface tension occurs at the interface between a liquid, a gas, and a solid. The surface of the liquid can be thought of as a membrane or a skin holding the liquid together (Munson, Young, Okiishi, 1998). Molecules on the exposed surface of the liquid have a net inward force of attraction to the other inner liquid molecules. This net inward force only occurs at the surface because inside the fluid, each molecule is completely surrounded by other molecules, balancing all of the intermolecular forces. Because a molecule on the surface only has forces of attraction from one or two sides, there is a net inward force. This net inward force is called surface tension and is illustrated in Figure 16.

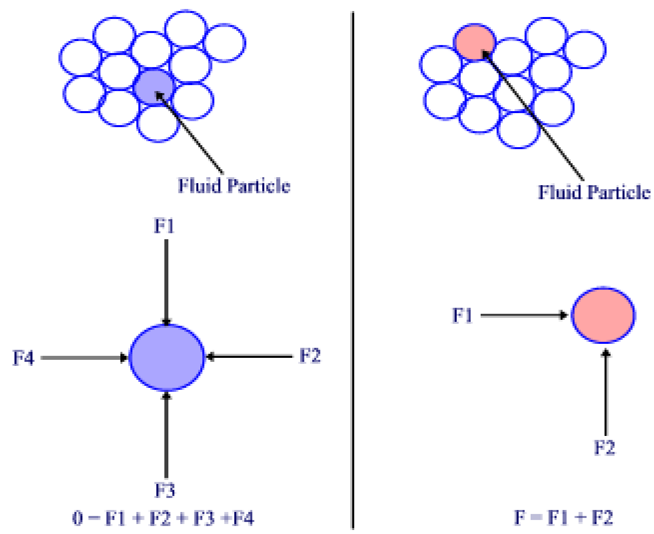


Figure 16: Pictorial Illustration of Surface Tension. The left view shows an internal particle and the right view shows a fluid particle on the perimeter.

One common example of surface tension at work is the bead of water on the hood of a freshly waxed car. The water forms small beads caused by the surface tension in the fluid. The forces of surface tension tend to pull the fluid into a tight ball. Mercury possesses the highest values of surface tension and mercury beads form almost perfect spheres when exposed to the atmosphere. This brings up the concept that different fluids, based upon their molecular properties, have different degrees of surface tension. The surface tension values for several common liquids are shown in Figure 17.

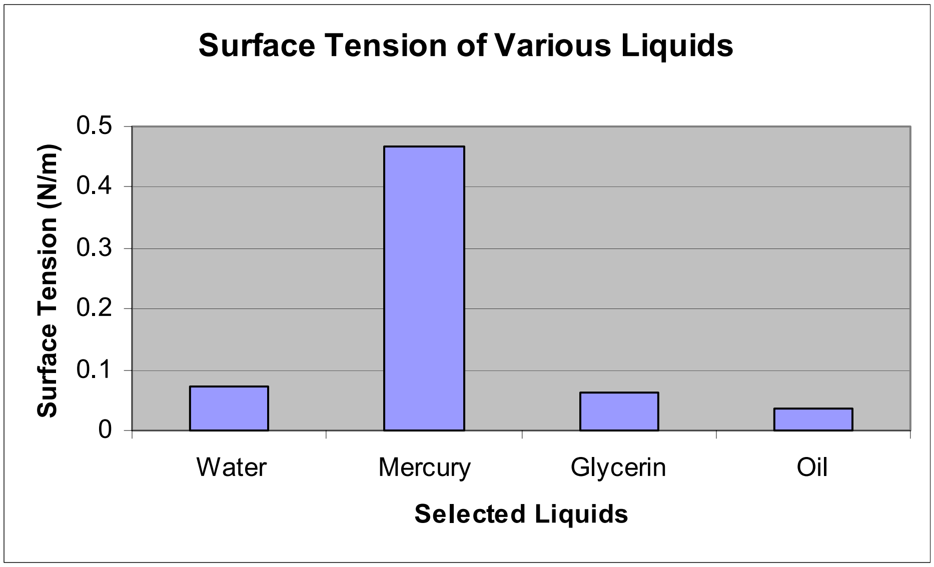


Figure 17: Surface Tension of Various Fluids. All of the surface tension values occur when the fluid in question is placed in contact with normal atmospheric air. The values for surface tension depend upon the medium of the surrounding fluid.

The capillary pumping increases with surface tension, therefore a high surface tension value leads to higher pressure heads produced by the wick. However there are several other important parameters that must be considered when selecting a working fluid. Most electronic packages have an operating temperature less than 100 degrees Celsius. This means that fluids like motor oil or glycerin, which boils at a much higher temperature are not practical. This also means that refrigerants like R-12 or R-134a cannot be used because they vaporize at a much lower temperature which would be below the temperature range desired for the operation of shipboard electronics. In addition the use of the refrigerants would require highly pressurized lines. Water is actually a good choice for the working fluid because its boiling point at one atmosphere closely mirrors the desired temperature range. Distilled de-ionized water is also a good fluid to use because it does not corrode or pollute the CPL system by leaving a residue or build up behind. Water is also non-toxic and easily found aboard a shipboard environment. However it is also important that the water is kept isolated from the electronics to prevent damage to the electronics. As long as the water is isolated in a closed, self-contained loop, its benefits can be harnessed and its phase change used to cool electronics.

3.0.2 CONTACT ANGLE

The next important parameter of the capillary pumping head equation is the contact angle. The contact angle is the angle that the meniscus of the fluid makes with the wall of the container or pore. The contact angle is purely a physical relationship and will depend upon the orientation of the fluid in the pore but also the orientation of the pore itself. The contact angle is not an independent property; it is constant for a given pore radius, geometry, and surface tension. Figure 18 illustrates how the contact angle is determined.

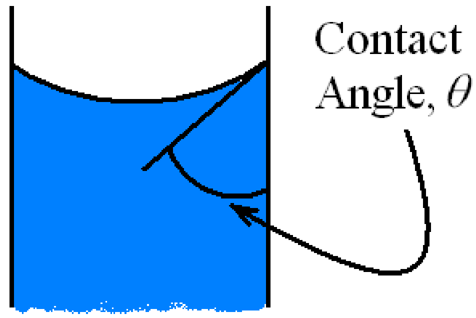


Figure 18: The contact angle is the angle that the meniscus makes with the wall of the container measured through the liquid. (Herron, 2001)

3.0.3 EFFECTIVE PORE RADIUS

Another important parameter in determining the capillary pumping pressure produced by a given wick and working fluid is the effective radius of the pores. As pore size in the wick increases, the capillary pressure produced decreases. For a CPL, the wick can either be freestanding or a sintered metal powder. The most effective range of pore size for CPL applications is between 10 and 30 µm. (Faghri, 1995) For a wick to be effective the pore radius must be within the desired range or a minimum pumping head will result. Figure 19 shows how changing the pore radius affects capillary pumping pressure, for a fixed contact angle and surface tension:

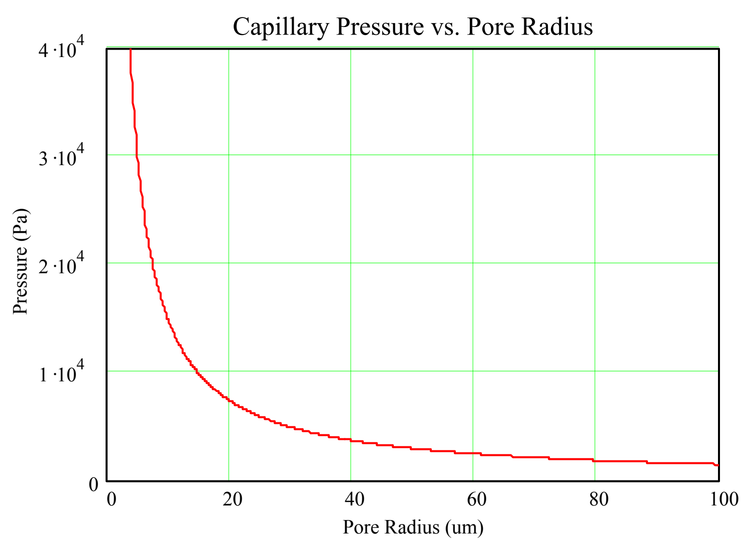


Figure 19: Capillary Pressure vs. Effective Pore Radius. The two are inversely proportional. As the pore radius increases, the pressure produced decreases dramatically. The contact angle is 0 and the surface tension value is 7.34 x 10^{-2} N/m. The working fluid for this plot is water.

The contact angle, surface tension, and effective pore radius only describe one small portion of the CPL system. There are several other aspects of CPL theory that will be applied to the CPL experimental loop.

For the current CAT evaporator and wick design, a maximum capillary pumping head of 9090 Pa was theoretically determined for the combination of the copper plate and copper wick. At optimum conditions, this is the largest pumping head that the system will be able to deliver based upon the wick alone.

3.1 LIMITATIONS OF CAPILLARY PUMPED LOOPS

There are several basic limitations that apply to capillary pumped loops. The first limitation is based on the rate of heat transfer in the condenser. The condenser limit determines the maximum amount of heat that can be removed from the loop. For internal flow a maximum fluid velocity of no more than 10 ft/s is considered standard. This fluid velocity determines the maximum volumetric flow rate through the condenser. The mass flow rate through the condenser directly affects the rate of heat transfer. It is related by equation 6, the same as for a heat pipe and is shown again below (Incropera, 2002):

$$Q_{out} = UA\Delta T \qquad \text{Eq. (6)}$$

In the cold plate, there are two separate surfaces for heat transfer, the tubes where the chill water flows and the tubes where the working fluid is cooled and condensed. Both of the heat transfer coefficients depend upon the flowrate of the water passing through them. To calculate the overall heat transfer coefficient a resistor model was used to create a single expression for the entire condenser. The overall heat transfer coefficient, UA, was calculated using actual data observed from the cold plate, and was found to vary between 550 and 560 W/K for the higher flowrates. This high overall heat transfer coefficient means that the condenser limit will not be reached during the course of study of the loop. The condenser is large enough to handle all of the desired heat loads used during this investigation.

Another limitation that the loop faces is the boiling limit, where the amount of heat applied to the flat plate evaporator is so great that it causes all of the fluid in the wick to evaporate before it can be replaced by fresh liquid. This limit can be mathematically calculated by the equation (Faghri, 1995):

$$Q_b = \frac{2A_vRT^2}{P_{sat}h_{fg}}\left(\frac{1}{r_b} - \frac{1}{r_w}\right)\left(\frac{t_v}{k_w} + \frac{t_w}{k_w}\right)^{-1} \qquad \text{Eq. (13)}$$

This boiling limit however does not consider the additional advantage of the gravity head from the thermosyphon characteristics. The boiling limit for a thermosyphon is determined by the equation (Faghri, 1995):

$$Q_{max} = 0.142(\rho_v)^{.5}(g\sigma(\rho_l - \rho_v))^{.25} \qquad \text{Eq. (14)}$$

The maximum allowable heat flux lies somewhere between the two. Due to safety concerns with the laboratory set up and testing procedure, the maximum experimental heat flux was not tested.

Several other limitations apply to the CPL loop in question, but they will be discussed in the fluid flow analysis. Before moving on to the fluid flow analysis there are two important factors that have a large effect upon CPL performance, subcooling and pitch and tilt. These concepts will be introduced below.

3.2. SUBCOOLING IN CPL LOOPS

Subcooling is an important parameter of two-phase heat transfer. Subcooling is said to occur when the working fluid is cooled to a temperature below its boiling point. In general, subcooling has a negative influence on CPL performance. When the subcooled water enters the evaporator plate, it must be reheated back to the boiling point during which it absorbs only 4.18 kJ/kg. Once the water begins to boil it absorbs 2257 kJ/kg in order to undergo the necessary change in phase. Subcooling can occur in the liquid return lines, flow restriction valves, and in the condenser itself. The effect of subcooling upon condenser performance is shown in Figure 20. As subcooling is increased, the performance of the cold plate decreases.

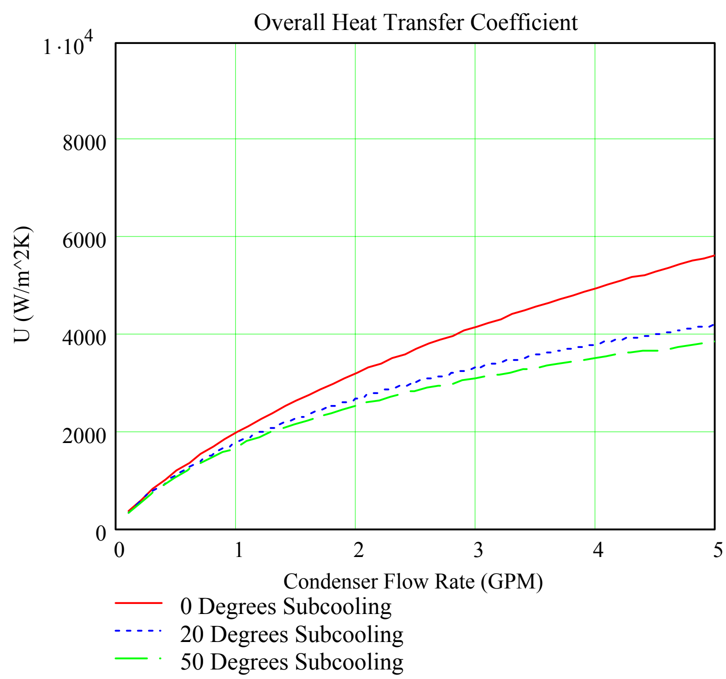


Figure 20: Subcooling and its effect upon the condenser. The subcooling is measured in degrees K

There are three curves in this plot: no subcooling, twenty degrees of subcooling and fifty degrees of subcooling. The overall heat transfer coefficient decreases as the subcooling increases because the mass flowrate of the system is reduced in order to account for the heat needed to be absorbed by the working fluid before it can vaporize and flow through the loop. As the overall

heat transfer coefficient increases, so does the condenser limit, and more heat can be removed from the evaporator plate. To achieve maximum condenser performance the amount of subcooling in the condenser must be limited.

One element of this project is to study and minimize the effects of subcooling on the CAT loop performance. In order to perform an accurate study of subcooling, the entire system must be well insulated. All of the vapor and liquid lines, along with the evaporator plate and cold plate, were insulated to prevent the loss of heat to the surrounding environment. Once the system is properly insulated, the amount of subcooling was controlled by the flowrate and temperature of the chill water through the cold plate. A systematic study of the subcooling was performed by varying the chill water flow rate and temperature and measuring the influence on the evaporator plate temperature.

3.3 TILT AND PITCH EFFECTS IN CAT LOOPS

Any system engineered to cool electronics at sea or in the air must be designed with stability and a consistent performance in mind. A successful device will have to overcome many different angles of pitch and tilt during its normal operation. Another experiment was performed to study the effects of tilt and pitch upon the CAT flat plate evaporator. The pitch of the evaporator plate is shown pictorially in Figure 21.

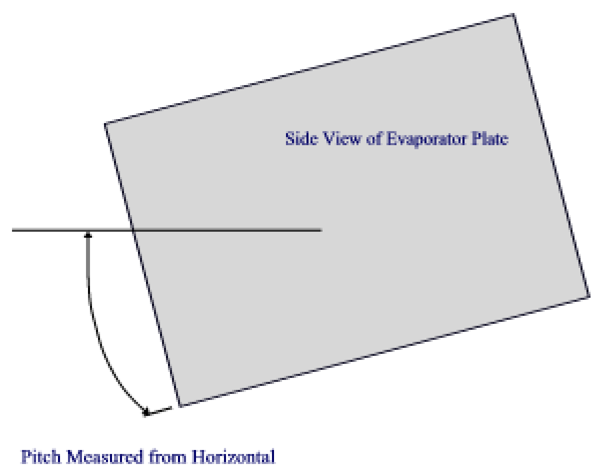


Figure 21: How the pitch of the evaporator plate is measured

The pitch represents the fore and aft angle of the evaporator plate itself. The initial or zero angle is when the edge of the flat plate evaporator is perfectly level. A positive pitch occurs as the front of the evaporator plate rises and a negative angle occurs as the forward end of the evaporator plate is dropped. This simulated the extreme angles that a ship at sea and aircraft during normal operation will experience.

The tilt represents the movement of the plate with respect to the vertical axis. The plate was tilted to the right and to the left. A zero angle will again occur when the plate is perfectly vertical. For a sign convention a positive tilt occurs when the evaporator plate is tilted to the right and negative to the right. A pictorial illustration of the tilt is in Figure 22.

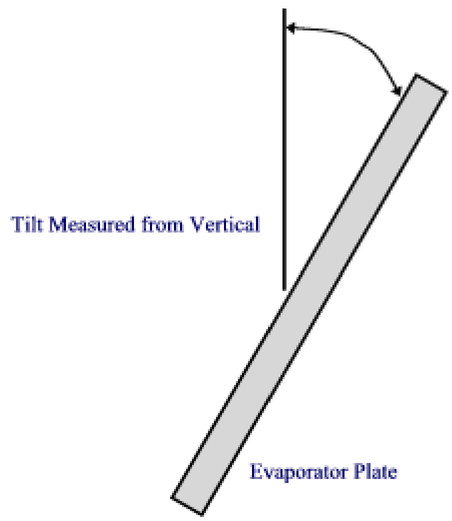


Figure 22: How the tilt angle of the evaporator plate will be measured

The combined pressure head created by the gravity head and the capillary wick should help maintain performance as the plate is tilted and pitched. It is important to note that for extreme tilt or pitch angles the fluid flow in the CAT loop might attempt to reverse itself, meaning that vapor will try to flow to the condenser through the liquid return line and liquid would travel from the condenser to the evaporator through the vapor return lines. If the CAT system attempted to switch directions during operation it would cause the flow to effectively stop and cease operation. This must be avoided, and thus the tilt and pitch were studied to see the maximum tilt and pitch that can be applied to the evaporator plate and still allow it to operate in the designed direction.

3.4 FLUID FLOW ANALYSIS

Fluid flows from high-pressure regions to low-pressure regions. In a CAT system this pressure difference is caused by two main factors, the capillary pumping head present in the wick and the thermosyphon effects due to the change in height between the evaporator and the condenser in the loop. The pressures throughout the system are related by the equation:

$$\Delta P_{CAP} + \Delta P_{GRAV} = \Delta P_{LIQ} + \Delta P_{VAP} + \Delta P_{FLOW\ LOSSES} \qquad \text{Eq. (15)}$$

Each pressure drop will be examined to see its effect upon the entire loop. If the sum of pressure drops is greater than the total pressure produced, no fluid will flow and the system will fail. This point can be experimentally determined by increasing the heat input applied to the evaporator

plate until the surface temperature starts to dramatically increase, meaning that dry out has occurred because the increased velocity has created pressure drops higher than the produced pressure.

The loop can be broken down into eight separate regions. An illustration of the eight different state points is shown in Figure 23. The state points are each referenced by a number. In this schematic, the fluid flows in a counterclockwise pattern starting from the flat plate evaporator. After the pictorial illustration, the points will then be plotted on a pressure vs. temperature saturation curve, shown in Figure 24.

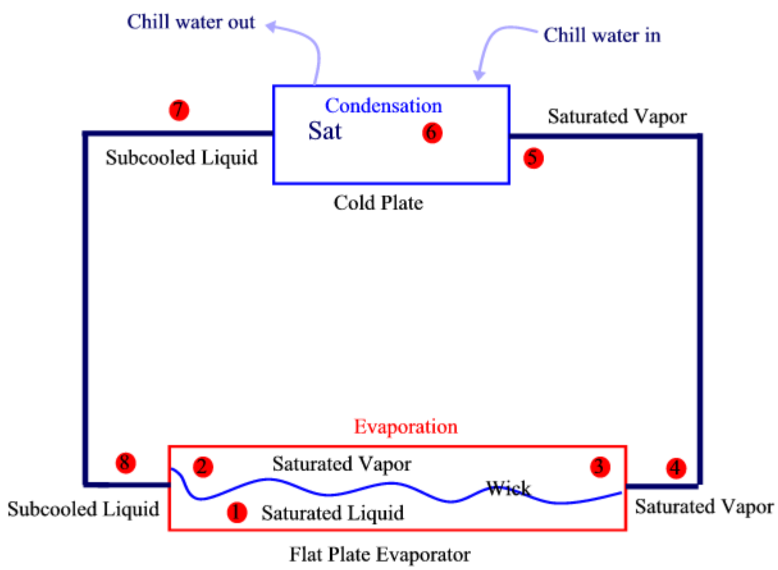


Figure 23: Loop Schematic for a CAT system.

State point 1 represents the saturated liquid in the evaporator, state point 2 is a saturated vapor, the pressure drop between state points 2 and 3 occurs due to vapor flow resistance in the wick and the vapor channels. The pressure drops between 3, 4, and 5 are all caused by flow losses due to vapor flow resistance caused by friction on the pipe walls of the system. State 5 represents the vapor entering the condenser. State point 6 represents the process of condensation as the vapor is condensed to a liquid. The fluid leaves the condenser at state point 7 as a slightly subcooled liquid and the pressure drop between 7 and 8 occurs as the liquid undergoes a pressure drop caused by flow losses. Once present in the evaporator as a liquid, the fluid returns to state point one and the cycle is repeated. The pressure drop between 8 and 1 indicates the liquid flow losses through the wick.

The saturation pressure vs. temperature curve with the eight state points is shown in Figure 24, followed by a pressure vs. temperature curve showing the same state points but this time with extensive subcooling, shown in Figure 25. The blue curve represents the saturation curve for liquid water.

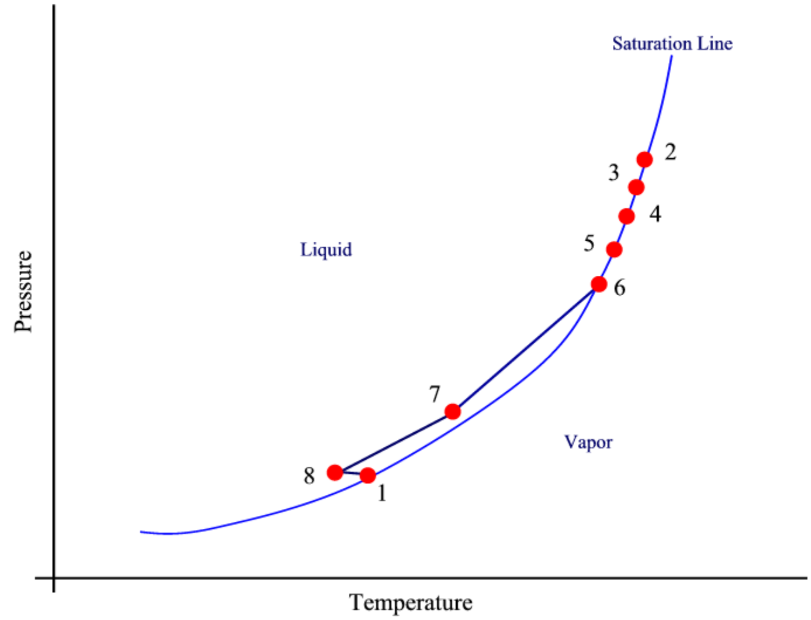


Figure 24: Minimum Subcooling CAT curve. This is the optimal curve for a CAT

If subcooling were to play a major role in the loop, it would push the state points away from the saturation curve. A heavily subcooled saturation curve showing the same state points is shown below:

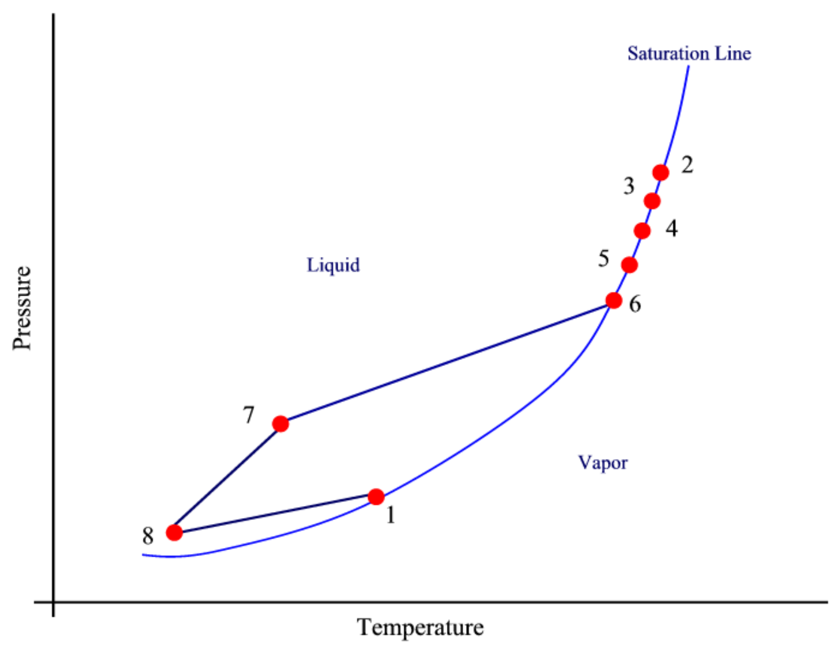


Figure 25: CAT System with extensive subcooling.

For the working fluid to evaporate it must be on the saturation line. Because the desired goal of a CAT system is two-phase heat transfer, the system must be kept as close to the saturation line as possible. This can be done by insulating the system and carefully controlling the flow rate of the chill water in the condenser.

3.4.1 CORRELATION BETWEEN HEAT INPUT AND FLUID FLOW

In the previous section, the pressure drops throughout the loop were qualitatively discussed. The pressure losses will now be quantitatively examined. As stated earlier, the total heat applied to the evaporator plate is directly related to the mass flow rate and the amount of subcooling present in the system. The combined equation is listed below:

$$Q_{in} = m_{dot}(h_{fg} + Cp(T_{sat} - T_{sub})) \qquad \text{Eq. (16)}$$

The heat input is largely dominated by the latent heat of vaporization, h_{fg}, which is two orders of magnitude greater than the specific heat of water, C_p.

The mass flow rate is in turn related to the pressure drops throughout the system and the specific geometry of the evaporator plate and fluid flow lines. By making the heat input a function of the mass flowrate, a total pressure head for the system can be predicted and a fluid velocity determined from the following equation. (Munson, 1998)

$$\Delta P = \frac{fL\rho V^2}{2D} + \frac{K_L V^2}{2g} + \gamma(h_2 - h_1) \qquad \text{Eq. (17)}$$

The above equation can be divided into three major parts; major losses, minor losses, and the change in height between the evaporator and the condenser. The major losses will be considered first. The major loss term is the first term in the above equation:

$$\frac{fL\rho V^2}{2D}$$

In this equation, f is the frictional factor based upon the velocity of the fluid and the roughness of the pipe itself. A good estimate of the frictional factor for smooth pipes is given by the Blasius relationship where the Reynolds Number, Re, of the fluid is greater than 2300. (Munson, 1998)

$$f = \frac{0.316}{Re^{.25}} \qquad \text{Eq. (18)}$$

The Reynolds number is the ratios of interial forces to viscous forces, and scales with the fluid velocity. The Reynolds number is the most important parameter in determining whether the fluid is flowing in a laminar or turbulent pattern. Fluid transitions from laminar to turbulent flow at a Reynolds number of about 2300. The Reynolds number depends upon the velocity of the fluid in question, V, the diameter, D, of the pipe, and the kinematic viscosity, υ, of the fluid. The equation for the Reynolds number of fluid in pipes is given below:

$$Re = \frac{VD}{\upsilon} \qquad \text{Eq. (19)}$$

Modeling the Reynolds number at different points throughout the loop shows that the flow will transition from laminar to turbulent when the heat input reaches 1500 W.

Completing the first term in Eq. (17) are both the length, L, and diameter, d, of the pipe in question. This term is known as the major loss term and counts for the largest proportion of the pressure drop caused by the flow of the liquid in the system. As the velocity increases, the major losses increase proportionally.

The second term in Eq. (17) comes from the minor losses. Minor losses occur as the fluid flows around a bend in the pipe or through a valve or another orifice. The most significant factor in the minor loss equation is the K_L factor that depends upon the pipe geometry. For example, the K_L value is 1.5 for a 90 degree bend in a pipe. (Munson, 1998) The minor losses in the loop can all be summed together as K_T, the total minor loss factor and then multiplied by the corresponding parts of the equation.

Finally, the last term in Eq. (17) is the pressure head created by a change in height. This is the term that accounts for the thermosyphon effects on the CPL equation. The change in height between the evaporator and the condenser is shown in Figure 26.

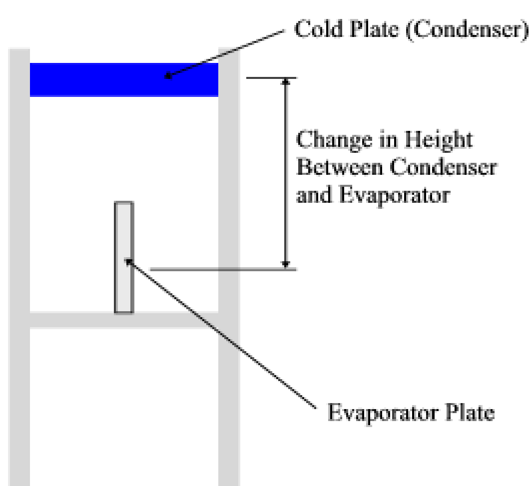


Figure 26: Thermosyphon effect on the CAT system

Placing the evaporator below the cold plate creates a positive pumping head as the condensed water falls back down to the elevation of the evaporator plate. This positive pumping head assists the capillary pumping head created by the wick.

3.5 PROPERTIES OF CPL WICKS

The wick is an important element in a capillary pumped loop. The capillary pumping head produced by the wick has been detailed in the beginning of this section. However, there are several other important properties of the wick that must be examined to see their individual effects upon the performance of the CAT loop system. For the capillary pumping head the most important parameter of the wick was r_{eff}, the effective pore radius of the wick. Most wicks used in CPL applications vary in pore radius from 10 μm to 40 μm. The effective pore radius is the mean radius of the pores comprising each wick. The thermal conductivity of the wick is another important parameter. Along the same lines, the porosity of the wick measures the amount of pores per unit volume. High porosity is desired to minimize the pressure drop through the wick. High thermal conductivity means that heat is easily transferred from the evaporator walls into the wick and then from the wick to the working fluid. The wick thickness is also important. The thicker the wick, the higher the pressure drop. This pressure drop works against the flow and lowers the pumping head produced by capillary action. The permeability of the wick is also important, the permeability is a measure of the wick's resistance to axial flow. (Faghri, 1995) A high permeability minimizes the losses present in the wick. All of these properties are important to the CPL system but most of them are conflicting. To achieve a small pore radius, porosity decreases along with permeability and thermal conductivity decreases as well because of the increased void space. The thickness of the wick is often a function of its composition, meaning that small pore radius wicks are usually thicker to keep the wick from falling apart. Therefore choosing a wick must achieve a balance among all of the different parameters.

There are two basic types of wicks that are available for use in CPL systems. The first are sintered metal wicks. Sintered metal wicks are made from metal powder; therefore these wicks tend to have a high thermal conductivity. The wick is constructed by placing metal powder on the desired surface, which is heated to upwards of 1000 degrees Celsius. Using this process, sintered metal wicks can achieve very small pore sizes with low porosity. The small pores can result in a high pressure drop as the working fluid is forced through the wick. One of the major disadvantages involving sintered metal wicks is that they are not freestanding. In other words, the wick is not able to support its own weight. The wick becomes an integral part of the surface and cannot be replaced or removed. This problem becomes important when corrosion becomes a factor. The second type of wick is made of a porous plastic material. The pore sizes are typically larger, which produces a lower pumping head. They also have a lower thermal conductivity because plastic is a thermal insulator. Most of the plastics also are hydrophobic and have to be specially treated for applications where the working fluid is water. One of the major advantages of these wicks is the fact that they are freestanding. They can be removed or interchanged easily based upon the desire of the designer or operator. The plastic wicks also tend to have a lower pressure drop through them as well as a higher porosity. A picture of a plastic wick is shown Figure 27.

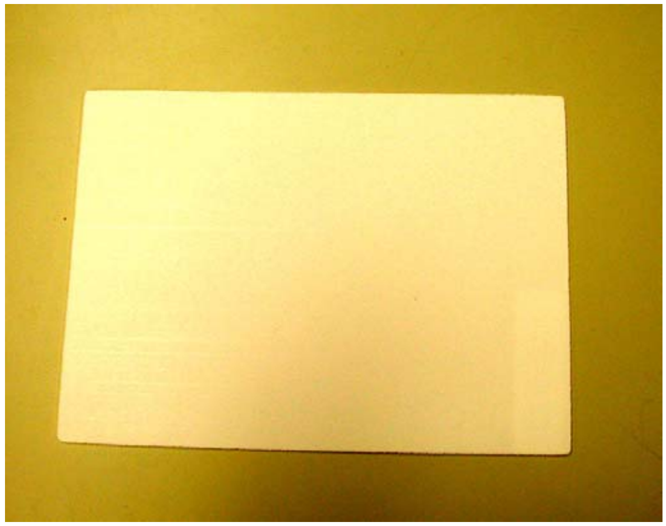


Figure 27: A 40 µm Plastic Porous Wick

Figure 28 depicts the sintered copper wick located in the top plate of the evaporator assembly.

Figure 28: A 16.15 µm Sintered Copper Wick

The copper wick is not freestanding but has a much smaller pore size and can thus produce a higher pumping head. The aluminum plate and plastic wick combination can theoretically produce a capillary pumping head of 3670 Pa and the copper version reaches 9090 Pa for this design. The copper wick produces a higher pumping head which should increase performance of the CAT loop.

For naval applications, the wick must also be low maintenance and be able to function for extended periods of time in order to be effective. If the wick must continually be replaced due to corrosion or fatigue then the CAT loop will not be effective as a heat transfer device. The plastic wick in the aluminum plate has a maximum melting temperature of 160 degrees C and

experiences softening around 130 degrees C. The normal operation of the plate is very close to the softening temperature and after the subcooling test was completed the wick experienced some melting and shrinkage. A picture is shown in Figure 29.

Figure 29: Aluminum Plate with Plastic Wick After Subcooling Tests

It is visibly apparent that the wick was warped by the internal temperature. Because of its deformation, its effectiveness over time is called into question. To melt the copper wick the temperature has to approach 1000 degrees C and thus the copper wick does not face the same deformation or melting restrictions. Thus longevity is also a major concern with the design and selection of the wick.

The wicks are an important factor in understanding capillary pumped loop theory and an important component in the CAT loop. Not only do they produce a pumping head they are also key in spreading out the liquid as it enters the body of the plate to maintain an even plate temperature and prevent local and plate dryout. The wicks will also help prevent channeling of the vapor and liquid into only a small portion of the plate.

3.6 CPL THEORY CONCLUSIONS

Having presented the background and genesis of the CAT loop, the physical elements and their design will now be examined. Without understanding basic CPL theory it would be difficult to understand the specifics behind the flat plate evaporator or the cold plate. The CPL theory introduced in this section will now be used to analyze and understand the experimental loop.

Typical capillary pumped loops have had their mass flow rates limited by the capillary pumping head, but by combining the characteristics of a thermosyphon, much higher mass flow rates can be obtained and help contribute to a more consistent performance under all conditions. The wick is not just responsible for the creation of a pumping head but also responsible for spreading the liquid water out over the entire surface of the plate. This will keep the entire surface of the plate at a near uniform temperature. The gravity head serves as the primary pumping head of the system and is key to cooling higher heat inputs than a simple CPL could. A regular CPL is limited by the pumping ability of the wick. For example, ENS Herron's horizontal flat plate evaporator / CPL combination could only remove 800 W of heat (Herron, 2001). The CAT loop is able to remove over 1600 W from both sides of the plate at the same time. Thus the CAT loop is a combination of the best characteristics of a CPL and a thermosyphon loop. Together the combination accounts for the weak points in each of the separate designs

4.0 DESIGN OF THE FLAT PLATE EVAPORATOR

The evaporator plate design was subject to several initial design constraints. To be able to cool Navy electronics, the flat plate must fit in Navy electronic cabinets and COTS circuit cards. The specified dimensions for the shell racks and mountings require the evaporator plate's overall dimensions to be twelve inches long by nine inches high. The evaporator must remove heat from both sides and be no wider than a half inch. There have been many horizontal flat plate designs but there are no vertical flat plate double-sided evaporator plates to date. Therefore the flat plate evaporator design contains several revolutionary aspects. The working fluid was also specified to be water because of its intended shipboard applications. In addition, the material of the evaporator plate and the material used in the wick have to have low electro-potentials because two dissimilar materials in contact with an ionizing fluid like water will cause the two materials to corrode, forming a battery and producing a voltage. This restricts the material of the wick to either an inert plastic or the same material as the evaporator plate.

4.1 MATERIAL SELECTION

Three main parameters were used to guide the materials selection for the evaporator plate. These parameters are a combination of the design requirements, strength of materials analysis, and a material property analysis. The first design parameter looked at was the thermal conductivity of different metals. This property was used to generate a list of suitable materials from which a buckling analysis could be made. The final parameter was the compatibility of the selected material to the working fluid of water. Material selection is a critical aspect to the design portion because of the revolutionary nature of the flat plate evaporator.

4.1.1 THERMAL CONDUCTIVITY

Due to the given design constraints, material selection was very important and thermal conductivity was used as a major limiting factor to reduce the number of materials considered. Thermal conductivity is a measure of how well a material transfers heat through conduction. In order to remove heat from electronics and transfer it to the working fluid, the thermal conductivity of the material selected has to be relativity high. A material with a low thermal conductivity would result in higher temperatures because of the higher resistance to conduction. To achieve operating temperatures less than 100 degrees C, materials with a high thermal conduction were examined. Several materials were considered for use and their thermal conductivities are shown in Figure 30.

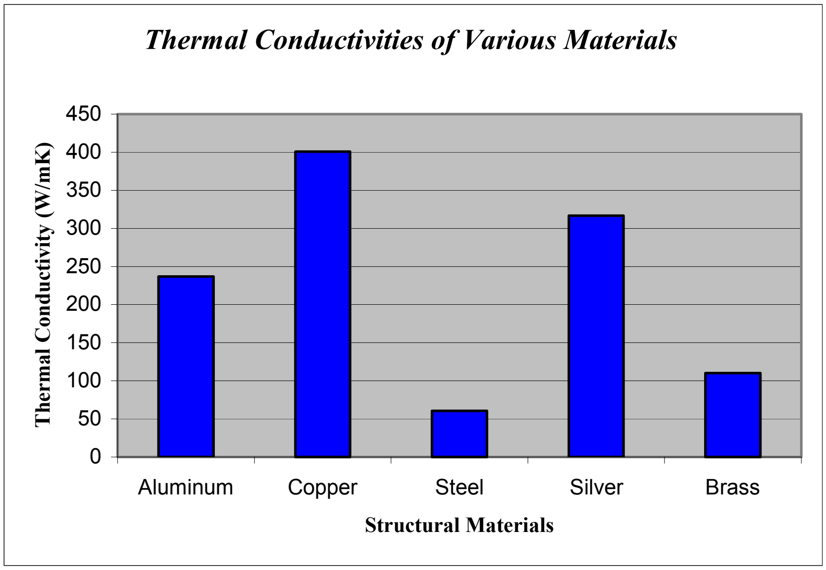


Figure 30: Thermal conductivities of various materials

After the thermal conductivity analysis, two materials emerged as likely candidates for design selection, copper and aluminum. The other materials will not be effective due to their cost, machineability or their weight.

4.1.2 BUCKLING ANALYSIS

The next step in the materials selection process was to perform a strength of materials analysis upon the loads expected to be endured by the plate during its normal operation. Because the evaporator plate cannot be thicker than a half inch; this made material selection all the more critical because if a material is chosen that cannot withstand the working loads, the plate will fail. A MathCAD worksheet was produced to evaluate the ability of each material to withstand the buckling pressure caused by one atmosphere of pressure on the exterior of the plate. The pressure upon each surface was modeled as a distributed load and then changed to a point load at the center of the plate. The results of this analysis are shown in Figure 31.

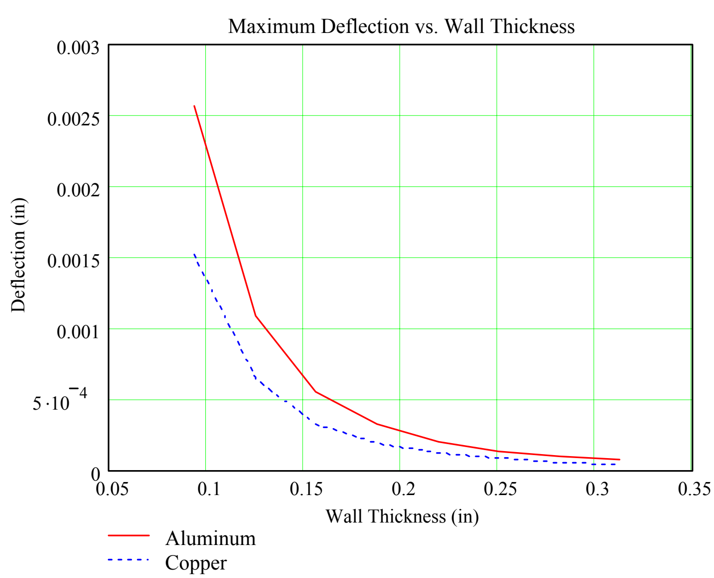


Figure 31: Deflection vs. Wall Thickness for Copper and Aluminum

The result of the buckling analysis shows that copper will deform less under stress than aluminum. This means that copper, with its high thermal conductivity would be the best choice to machine for the evaporator plate.

4.1.3 WORKING FLUID COMPATIBILITY

The final check upon materials selection was to analyze the effect of the working fluid upon the selected material for the evaporator plate. This was an important check to make because it will prevent corrosion from being a major factor affecting the performance of the evaporator plate. After satisfying the first two criterion, the material of the evaporator plate must also be able to withstand the working fluid without extensive corrosion due to the working fluid, water. If the plate material was chosen without regard to corrosion, the evaporator's performance will degrade over time. For use by the Navy, the material used in the flat plate evaporator must be relativity resistant to corrosion to ensure a long lifetime.

The best material to use as the basis for the flat plate evaporator was copper. When aluminum is combined with water, the creation of non-condensable gasses occurs and this effect will reduce the effectiveness of the aluminum plate over time. (Faghri, 1995) Thus the copper satisfies the major design criterion and has the highest thermal conductivity of any commercially available metal. An aluminum plate was also be constructed as a prototype because of the expensive nature of the copper material. The aluminum plate was used for several of the major tests before being replaced by the copper plate.

4.2 PLATE GEOMETRY

Typical flat plate evaporators used in loop heat pipes and CPLs have been horizontal in orientation. The liquid is returned from below to the evaporator plate where it is then pumped up through the wick by capillary action. The heat input causes the liquid in the wick to evaporate, the vapor then is collected and travels to the condenser and the process is repeated. The capillary

pumping head produced by the wick forces the water up through the evaporator plate in response to the heat input. A cross sectional view of the characteristic features in a horizontal flat plate evaporator is shown in Figure 32.

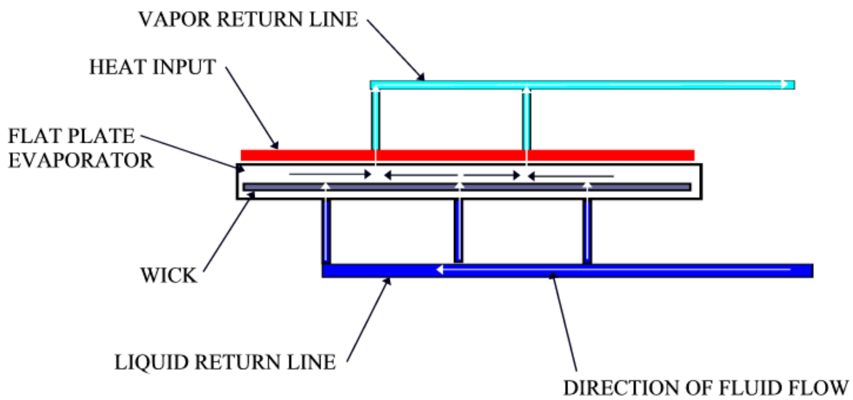


Figure 32: Typical Flat Plate Evaporator Geometry

The working fluid is pulled up from underneath the plate by capillary action caused by the wick. The liquid is then evaporated and travels upward and out of the evaporator plate using the vapor transport lines. The heat is only input at the top surface of the evaporator plate. This geometry is typical of horizontal flat plate evaporators including ENS Herron's from his 2001 Trident project. The problem with horizontal evaporator plates and electronics cooling lies in the fact that most electronic stacks are vertical in nature. This means that a totally new geometry must be created.

4.2.1 VERTICAL FLAT PLATE EVAPORATOR BASIC GEOMETRY

A vertical double-sided flat plate evaporator is a new concept and before the design could be created the internal geometry had to be determined. Because the cold plate is located above the evaporator plate, the main mode of operation for a vertical evaporator will be that of a thermosyphon, liquid cannot flow up, so the liquid input must be at the top of the evaporator plate. Because heat is going to be added on both sides, two wicks are needed for the evaporator plate to function properly. Wicking both sides of the plate will serve to increase the pumping head and also spread the liquid water out over the entire surface of the plate walls ensuring an even temperature distribution. Without the two wicks, there would be a large temperature gradient on both sides of the plate. Having two wicks also helps the plate maintain an even temperature as it undergoes tilt and pitch. The wick again ensures that the water will be evenly spread throughout the plate no matter what the basic orientation. The challenge with the wicks becomes buckling because the wicks are not self-supporting. This means that the load must be carried by the basic frame of the evaporator plate. The basic design geometry is shown below in Figure 33:

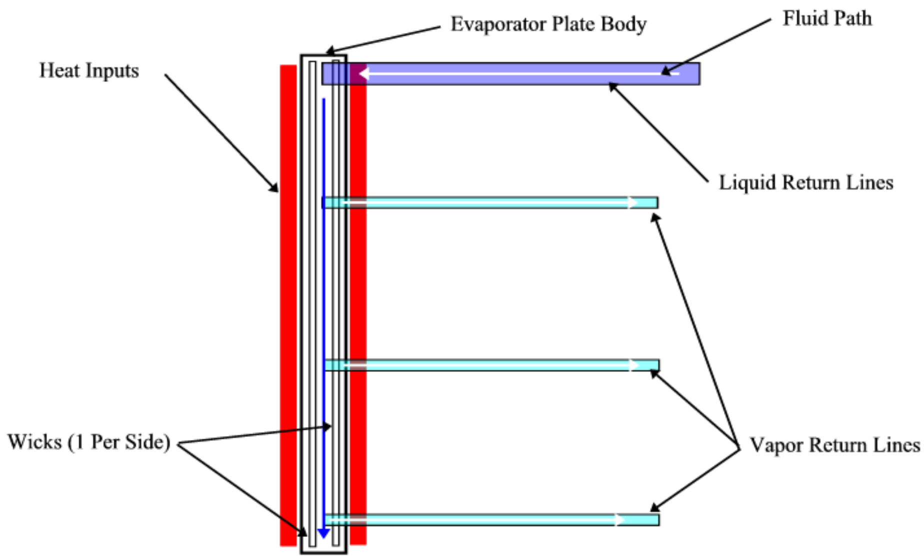


Figure 33: Vertical Flat Plate Basic Geometry

The large surface area of both sides of the evaporator plate will readily allow heat to be transferred from the heaters to the fluid inside the plate, but combined with the fact that sides of the plate must be separated by vapor space in order to allow room for evaporation, calls the structural integrity of the plate into question even after deciding upon the use of copper for the evaporator plate materials. To prevent buckling and allow heat to flow from one side of the plate to the other, semicircular ribs were designed to fit inside the evaporator plate.

4.2.2 ADDITION OF SEMICIRCULAR SUPPORTS

The plate will be operating at up to one atmosphere of pressure during normal operation. Because of the desired wall thickness and overall plate dimensions, buckling was a major design question. To ensure that the plate would not buckle at less than one atmosphere of pressure, semicircular supports were designed into the bottom of one of the sides of the flat plate evaporator. These circular supports have three functions. They serve first to prevent the plate from buckling and deforming under one atmosphere of pressure. Secondly, they also serve as heat spreaders because they are in thermal contact with each side of the evaporator plate. Finally, their circular nature aids in the distribution of the liquid water as it drops down the sides the evaporator plate walls. The semicircular ribs are shown in Figure 34.

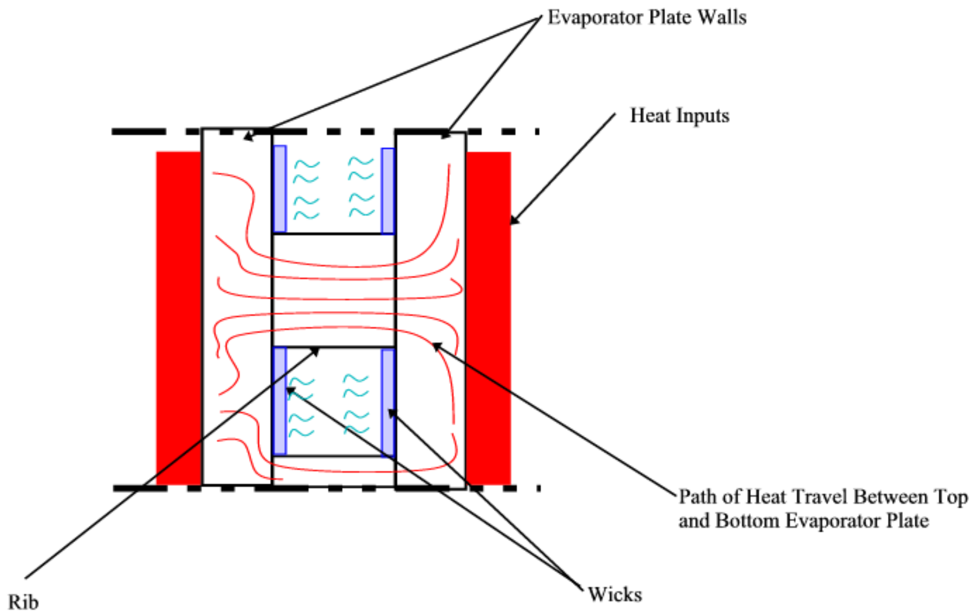


Figure 34: Rib Acting as Structural Support and Heat Spreader

These ribs allow the plate to transfer heat from one hot spot on one side to the cooler side. When the plate is tilted and pitched, the water will tend to flow to the lower side. Having heat spreaders in the plate will allow heat from the higher side of the plate to be transferred by conduction into the lower plate, maintaining a uniform temperature over the surface of the plate. Finally, the semi-circular shape of the ribs will cause the liquid water, dripping down through the middle of the plate to be spread out evenly over the entire surface of the wick. If these supports were not present the water would flow to the bottom of the plate and then be wicked away to the surface of the plate. Having the supports evenly spaced throughout the plate should aid in creating a uniformly wetted wick in addition to creating paths for conduction. In addition, the contact areas between the two wicks and the supports should become key spots for capillary action to take place.

4.2.3 LOCATION OF LIQUID RETURN LINE AND VAPOR RETURN PORTS

Another important design decision was to determine the location of the liquid and vapor lines into and out of the flat plate evaporator. Because the evaporator plate is designed to fit in an electronics cabinet, the piping must all be on one side of the plate for ease of installation and maintenance. Ideally, the liquid return line should enter through the center of the top of the plate. It would then allow fluid to flow equally in both directions and then down into the evaporator plate. Because the vapor transport lines are on the same side of the plate, channeling becomes a major design concern. Channeling means that the capillary action and heat transfer will occur on only a small section of the plate itself.

To avoid channeling at high heat inputs, the vapor transport lines should have outlets on both sides of the evaporator plate. Therefore the design constraints have created two major issues that must be taken into account during the design of the evaporator plate.

4.2.4 SEALING EVAPORATOR PLATE TO HOLD A VACUUM

Because the CAT system will be operating below one atmosphere, the plate must be able to prevent the outside air from penetrating the evaporator plate. Any connections and opening to the plate must be designed to prevent air from leaking in. To ensure that the plate would hold a vacuum, a lip was designed around the perimeter of the evaporator plate to act as a guide to help seal the two halves of the plate together. Instead of a typical assembly with fasteners, an epoxy was used to minimize the openings in the plate. Welding was another technique that could have been used, but once the plate was welded together it could never be taken apart. For this initial experimental application, the plate assembly was designed to be taken apart to examine the inside. For the actual mounting into cabinets, the evaporator plates would be welded shut. Therefore epoxy around the lip of the aluminum plate was determined to be a method of enabling the plate to maintain a vacuum and prevent air from leaking in during operation.

The epoxy was used on the aluminum plate and was found to be marginally effective. The combination of high temperatures and pressures turned the epoxy into a brittle material that simply cracked away. The epoxy held an inadequate seal and it is not an ideal solution, as it tended to leak over time. For the copper plates, a new solution of using a silicon gasket material, RTV, combined with bolts was implemented. The plate design was modified slightly to leave space for the bolt holes and the lip was reduced to allow for the gasket. This solution was much more effective and it is relatively independent of pressure and temperature, ensuring that the seal will last for a long time.

4.3 EVAPORATOR DESIGN

The evaporator plate was divided into two separate plates to ensure manufacturability and assembly. There are a top and a bottom plate to the evaporator assembly. The bottom plate was designed to contain all of the complicated geometry and plumbing connections while the top plate acted as a cover plate and fit over all of the heat spreaders and interior geometry. The bottom plate design became the most critical part of the entire design process.

4.3.1 BOTTOM PLATE DESIGN

The bottom plate was initially designed using AutoCAD. The design goals were to create a vertical flat plate evaporator that will fit in a Navy electronics cabinet. These dimensions constrained the outer appearance of the plate itself. The inner geometry was controlled by the fact that the liquid return lines had to be located above the vapor transport lines to take advantage of the gravity head created by the falling liquid. The constraints for the heat spreader are derived from the buckling analysis performed on the evaporator plate. After the initial design was created, it was revised several times. The heat spreaders were initially triangular in form to aid in the machinability of the evaporator plate but after consultation with the Naval Academy's

Technical Support Department, they were changed to semicircular supports. The drawings for the evaporator plates are listed in Appendix 1 that contains all of the design drawings used during the construction of the plate and the rest of the loop. The dimensions for the sealing lip were also modified several times to ensure that the plate would retain a vacuum. After the design was finalized, a solid model was created using the IDEAS solid modeling program. A picture of the bottom plate from the solid modeling software is shown in Figure 35.

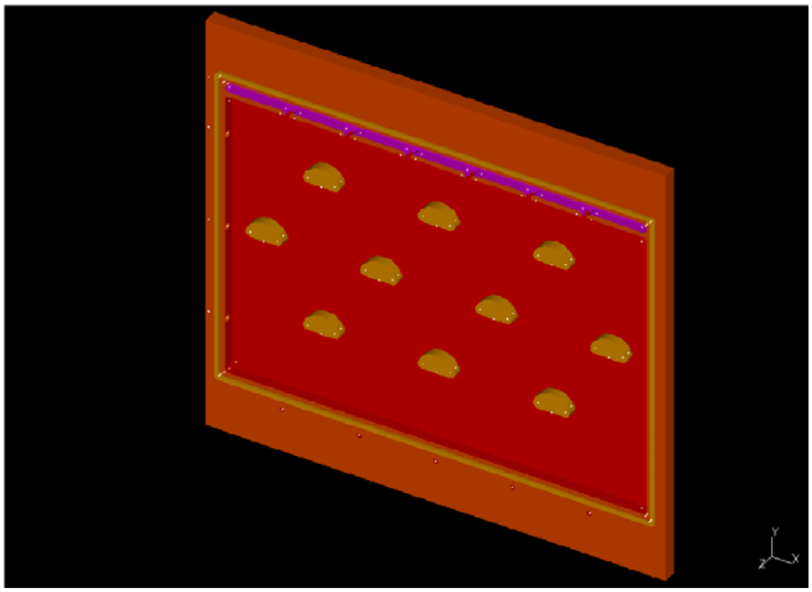


Figure 35: Solid Model of Evaporator Baseplate. This model was created in IDEAS.

After the solid model was created, the plate needed to be machined and tooling paths were created using the IDEAS software for the Haas Computer Numerically Controlled (CNC) machine in the machine shop. The tooling paths contain all of the information necessary for the CNC machine to manufacture the plate. This was the only way to machine the complicated geometry in the bottom plate. It would be extremely difficult and time consuming to machine the bottom plate in a manual mill operating only upon operator inputs. There are almost 10,000 lines of machine code present in the tooling path file that was loaded into the mill. The CNC machine was then loaded with the tooling code and used to machine both sides of the evaporator plate.

After the tooling paths were created, a prototype evaporator plate was manufactured using foam insulation to debug the tooling paths and check to ensure that the two halves of the evaporator plate fit together. The errors generated in the tooling path were then corrected and the first set of aluminum plates was manufactured. A picture of the aluminum bottom plate is shown in Figure 36:

Figure 36: Aluminum Evaporator Baseplate

4.3.2 TOP PLATE DESIGN

The design for the top plate was much simpler. The major issues with the actual design of the top plate involved machining a groove with the proper clearance to allow a vacuum tight seal. The same design process was followed with the top plate as was with the bottom plate. The original design was created in AutoCAD before a solid model was constructed in IDEAS.

From the solid model, the tooling paths were generated. A foam prototype was made of the top plate as well, and the fit of the male and female lips were checked to ensure the accuracy of the solid models. Then the aluminum top plate was machined. A picture of the top and bottom plate is shown in Figure 37.

Figure 37: Evaporator Top and Bottom Plate before assembly

4.4 INSTALLATION OF THE WICKS

The plastic wicks were measured and then cut out of the porex 40 μm material and glued to the surfaces of the aluminum plates. The wicks were placed flat in contact with the top and bottom plate and epoxy was applied at the four corners of the wick. The epoxy was then allowed to dry before the plates were assembled.

The copper wick had a much more involved process. The copper plates were mailed to Bechtel-Bettis Inc. in Pittsburgh, PA and the copper powder was sintered upon both sides of the copper plates. The copper powder was laid into the plate at a 1/16th of an inch thickness and then the plate and powder combination was heated up to 1000 degrees C in an air free chamber. The absence of air prevented the wick from oxidizing in the extreme temperature. After the wick was sintered in place it was then returned to the Naval Academy and prepared for testing. A picture of the copper bottom plate with the sintered copper wick is shown in Figure 38.

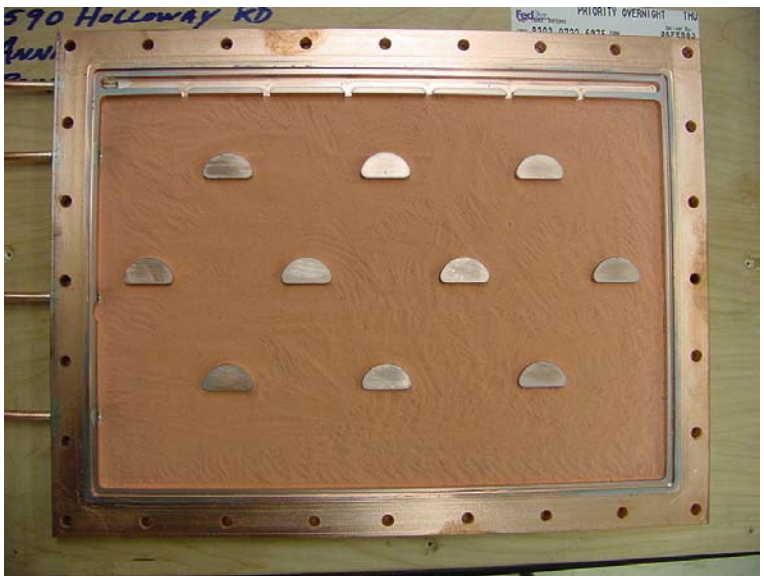


Figure 38: Copper Plate with Sintered Copper Wick

5.0 LABORATORY CONFIGURATION AND SET UP

The flat plate evaporator is just one of the many components of the capillary assisted thermosyphon loop. Other important pieces include the cold plate, the data acquisition unit, the piping, the insulation, and the instrumentation. All of the different components in Figure 39 will be discussed in detail in this section.

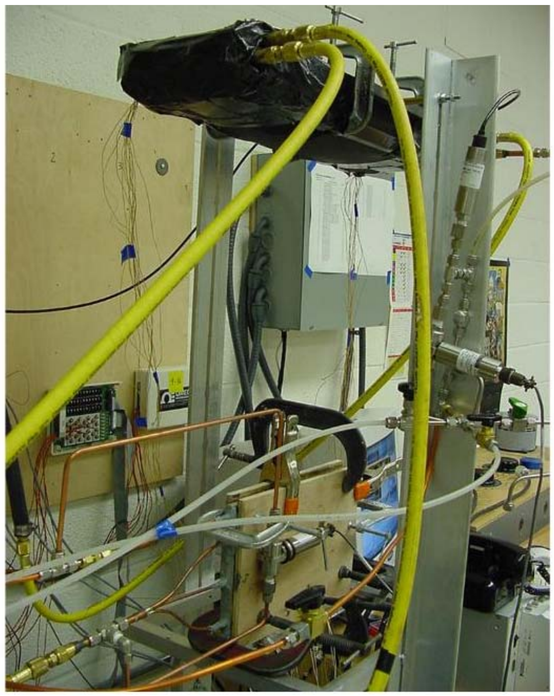


Figure 39: Basic Laboratory Setup. The cold plate is under the black insulation at the top of the picture. The evaporator plate is located near the bottom.

The theory in the previous sections all describes the relationship between the evaporator and the cold plate and how the fluid flows from one section to another. Figure 39 shows the location of these two key components. As stated previously, the cold plate is mounted above the evaporator plate to create a positive gravity head throughout the system. The height of the evaporator and condenser was not changed during the experiment. Instead, a pressure restriction valve was installed in the system and used to mimic changes in height or to simulate other flow losses and pressure drops. To begin the discussion of the other elements in the loop, the cold plate, which acts as the condenser for the loop, will be examined first.

5.1 THE COLD PLATE

The condenser is one of the most important elements of a CPL system. As stated earlier it transfers the heat from the working fluid to chill water. Most shipboard condensers are shell and tube heat exchangers similar to the one shown in Figure 40.

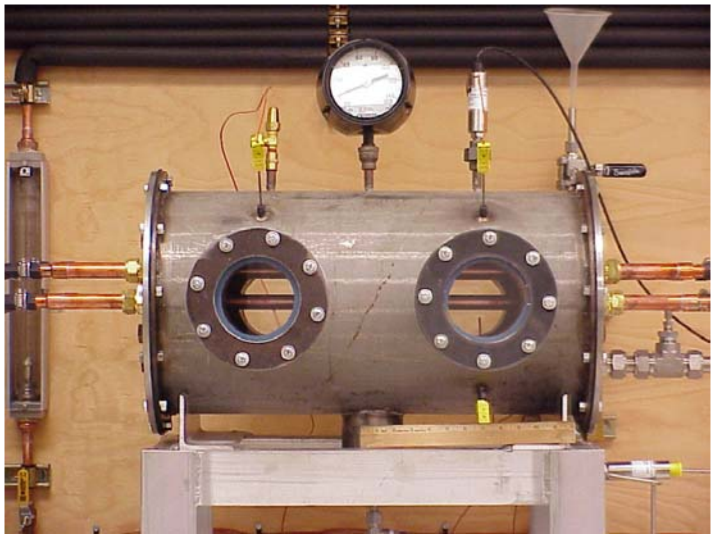


Figure 40: Typical Shipboard Shell and Tube Condenser

These large condensers are very robust and almost independent of pitch and tilt. The vapor enters at the top of the heat exchanger and then passes over chill water lines where it condenses and is collected at the bottom and piped away. Large condensers such as these place a high degree of subcooling upon the liquid because their large size and surface area allows them to remove a large amount of heat. Many of these large shell and tube condensers are over engineered for their applications and can be replaced by smaller, more efficient heat exchangers. Compact heat exchangers have several advantages over the large-scale shell and tube condensers. The most important consideration is the thermal mass of the heat exchanger that dictates the time required for the system to reach steady state operation.

The compact heat exchanger is much smaller than its larger cousin and is designed in such a way that the subcooling can be controlled by the rate of the chill water entering the heat exchanger itself. One version of an effective compact heat exchanger is called a cold plate. In a cold plate, the chill water and working fluid lines pass parallel to one another in small diameter tubes secured in a thin plate. Instead of condensing and dripping down to the bottom of the condenser the working fluid is condensed right in its enclosed tube and piped back to the evaporator plate. To increase the cold plate’s effectiveness it can be configured as a cross flow heat exchanger where the working fluid and the chill water flow in opposite directions. A cross section of the cold plate used as part of the CAT loop is shown in Figure 41:

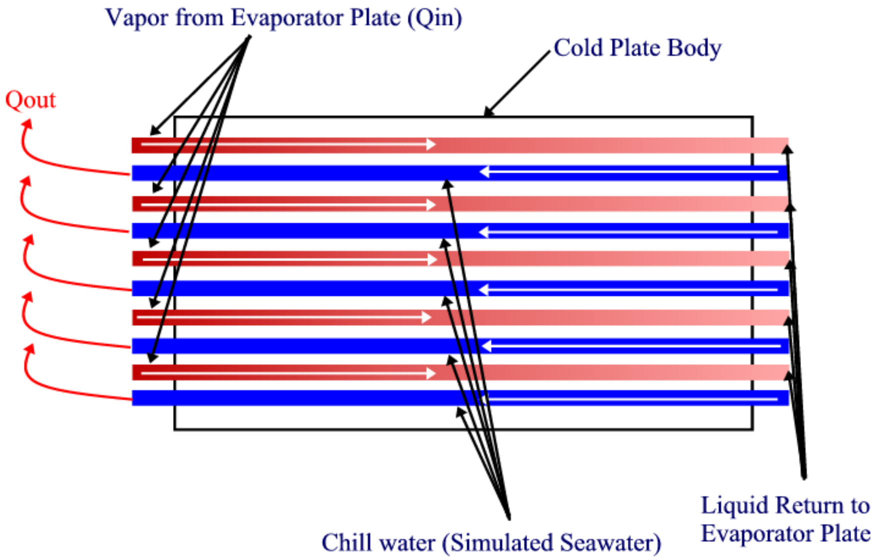


Figure 41: Cold Plate Cross Sectional View

There are several other advantages to using a compact heat exchanger like a cold plate to remove heat from the system. Compact heat exchangers are excellent for condensation applications where one fluid is a liquid and the other is a gas because of the high surface area per unit volume (Incropera, 2002). Cold plates have excellent longitudinal heat transfer across the width between the chill water and working fluid tubes. This capacity directly translates to a very high overall heat transfer coefficient (Incropera, 2002). Cold plates have less thermal mass than a large shell and tube condenser and thus reach steady state sooner. However, there are also several limitations to using cold plates including lower flow rate and less overall surface area than a large-scale shell and tube condenser. The lower flow rate means that the cold plate in a loop will reach the condenser limit much faster. The cold plate used in this experimental CAT system is shown below in Figure 42.

Figure 42: Cold plate before installation into loop

This cold plate has 3/8 inch diameter lines leading into and out of the plate which are connected to a manifold of six ¼ inch copper tubes which travel the length of the plate. The chill water flows through one set of tubes. The vapor from the evaporator enters the second set of tubes where the condensation process takes place as the vapor is turned back into a liquid as heat is removed by the chill water in the neighboring tubes. The maximum overall heat transfer coefficient for the cold plate was determined to be 5230 W/m^2K at a maximum flowrate of 9.18 GPM.

The cold plate is insulated using black foam rubber to limit the amount of uncertainty in the experiment. By insulating the condenser, losses to the environment are prevented, which made the energy balances easier to create.. A compact heat exchanger brings more control to the experiment than a much larger standard condenser. A picture of the insulated cold plate is shown in Figure 43.

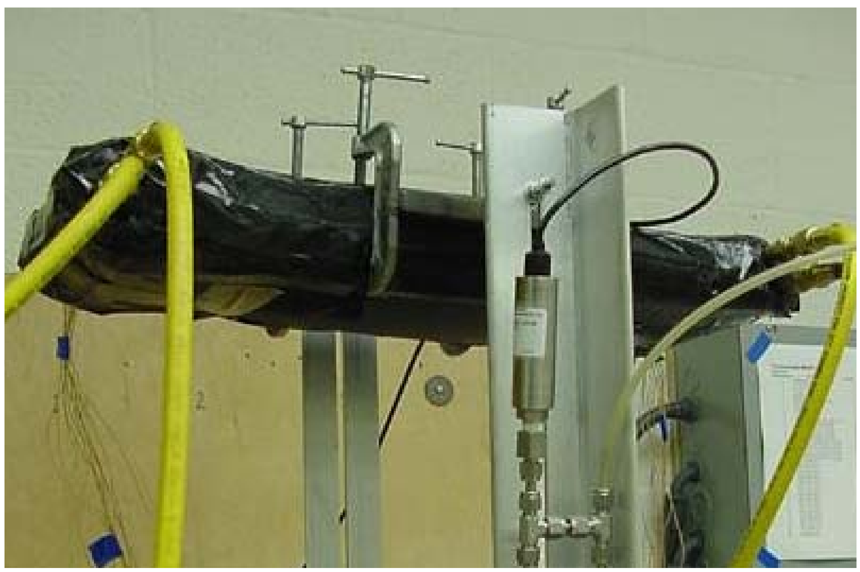


Figure 43: Insulated cold plate

5.2 THE PITCH AND TILT DEVICE

Another important part of the loop and the experimental setup is the tilt and pitch device. In order to test the influence of tilt and pitch, a fixture needed to be designed to allow the evaporator plate to have two degrees of freedom. The pitch and tilt device must also support the weight of the evaporator assembly. The tilt and pitch device was designed using aluminum because of its availability, strength, and ease of machinability. The device that was designed for this application is shown below in Figure 44.

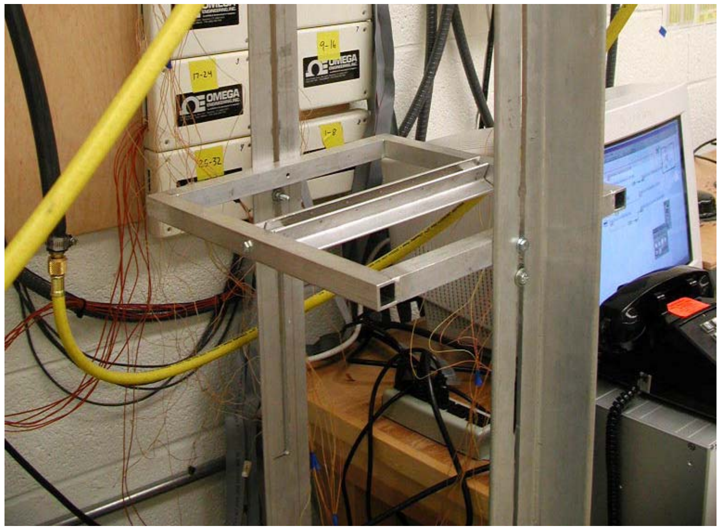


Figure 44: Tilt and Pitch Device. The device is located in the bottom half of the picture. This picture was taken before the evaporator plate was added.

The pitch and tilt device was mounted to the testing frame on both ends. The outer loop controls the pitch of the evaporator plate, while the tilt is controlled at the center where two pieces of aluminum angle have been welded together to form the mounting bracket for the evaporator plate. This mounting bracket is pinned along the centerline of the outer loop to give the device the desired two degrees of freedom. The plate assembly was designed to fit in the mounting bracket securely. The pitch and tilt device fits in grooves machined along the edges of the test stand which allow for the device to slide up and down to control the thermosyphon aspects of the experiment as well. The positive pressure head can be reduced by moving the pitch and tilt device up towards the cold plate or increased by increasing the distance between the two. All of the dimensions and drawings used to design the tilt and pitch device are displayed in Appendix 1.

5.3 LOOP SCHEMATICS

Now that the evaporator plate and the cold plate have been added to the CAT loop the system is complete. The two plates were then connected by a combination of copper tubing, plastic tubing, and refrigeration hoses. The copper lines were used in instances where the lines could be rigid and the plastic tubing was used to facilitate the tilt and pitch of the evaporator plate. A schematic of the entire loop is shown below in Figure 45. The schematic shows all of the major parts to the CAT loop along with a legend to detail the different symbols used in the loop.

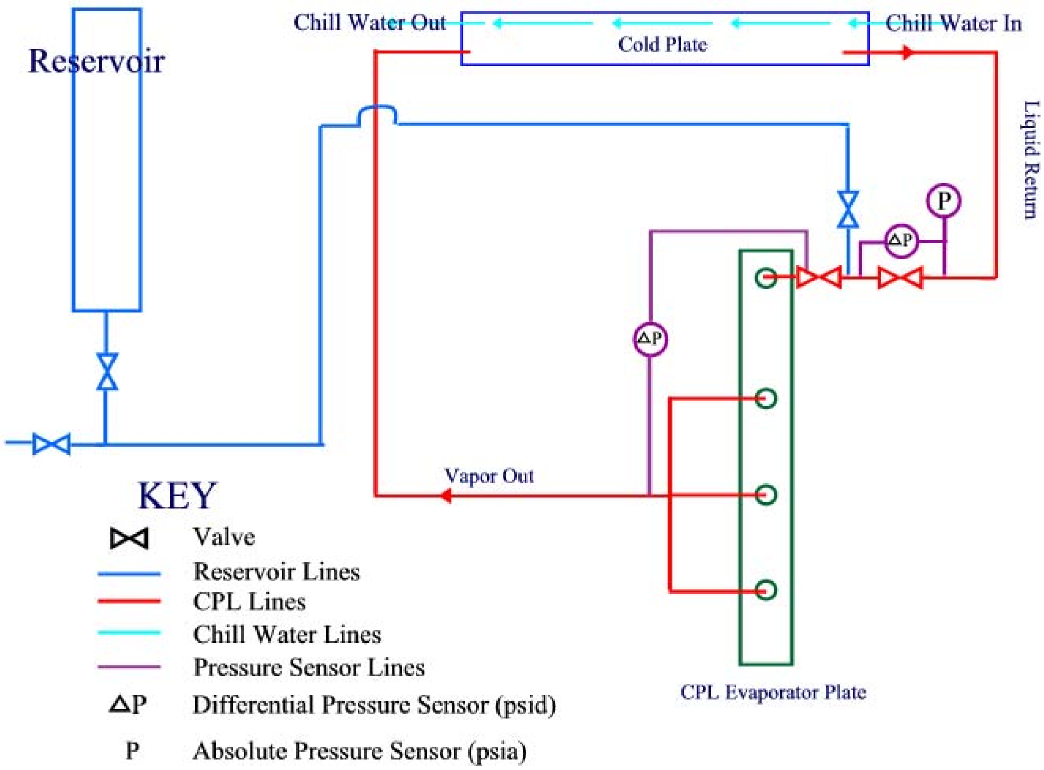


Figure 45: Loop Schematic

On this loop schematic there are actually three separate systems. The first step in the operation of the CAT is to pull a vacuum on the system using a vacuum pump. This vacuum removes all of the air and water vapor from the loop. The vacuum line is not shown on this plot for simplicity. The first component displayed is the fill tank from which the working fluid is added. Initially all of the working fluid is stored in the reservoir. The reservoir's lines are shown in a light blue to denote the path the water initially travels as it enters the system. After the vacuum has been pulled, the vacuum line is closed and the desired amount of working fluid is added from the reservoir. After the desired amount of fluid has been introduced into the system, the reservoir lines are closed. Next, the red lines in the schematic denote the actual CAT lines where the working fluid travels in the loop. The red valve is a needle valve, which was used for the pressure restriction test. The needle valve can be gradually closed to increase the pressure drop across it, which simulated the increased flow losses from friction. As stated earlier, the water is boiled in the evaporator by applying heat and condensed in the cold plate. The 3rd and final system is the chill water loop that will be discussed in the next section. The chill water loop is responsible for pulling the heat from the cold plate. The chill water loop must maintain the chill water at a constant temperature and flowrate during experimentation.

The loop schematic above also shows the location of the pressure sensors in the loop. The pressure sensors in the system are shown in purple and the principles behind their operation will also be discussed later in the instrumentation section. The pressure sensors in the loop are two

types, one that registers a pressure differential between two points and the other records an absolute pressure.

5.3.1 CHILL WATER LOOP

The chill water loop is separate from the CAT. The chill water is used to remove the heat from the working fluid inside the CAT. This transfer of heat occurs in the cold plate. The cold plate along with the chill water flow lines leading to and from the isothermal control tank are insulated. Therefore the heat gained by the chill water should the same as the heat deposited into the evaporator plate. The cold plate is the interface between the two separate independent loops. The chill water flows opposite the heated water to maximize the heat transfer capability of the cold plate. A schematic of the cooling water loop is shown below as Figure 46.

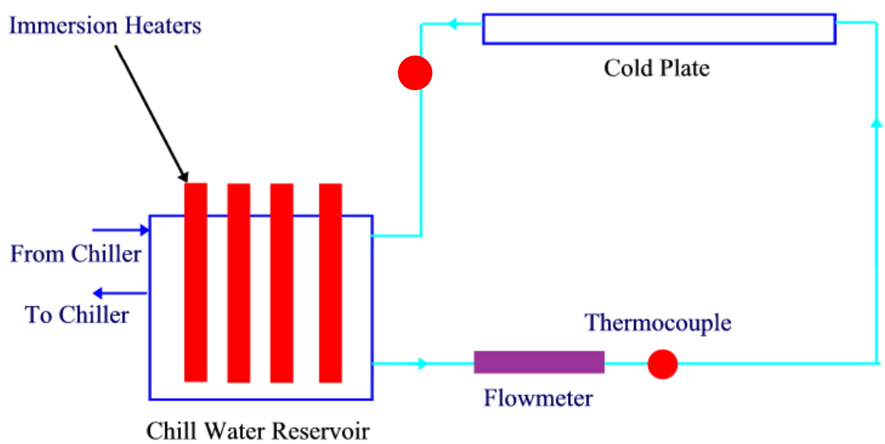


Figure 46: Chill Water Loop

The flow meter measures the volumetric flowrate of the water leaving the condenser. The thermocouple at the flowmeter is used to measure the temperature of the water. This temperature was used to derive the fluid properties of the chill water before entering the cold plate. The cold plate was also instrumented to monitor the change in temperature of the chill water across the plate, by a second thermocouple located in the chill water line after the cold plate.

The temperature of the chill water in the reservoir can be controlled and was varied during the course of investigation to simulate different ocean temperatures. The chill water can be cooled by a refrigeration unit that is connected to the reservoir, or heated by immersion heater placed inside the chill water reservoir to achieve and maintain the desired temperature for testing. The flexibility of the chill water loop is key to simulating the different environments that the electronics aboard a warship might be subjected to. The chill water loop allows the simulation of any seawater temperature in the world.

5.4 INSTRUMENTATION OF THE LOOP

The instrumentation of the entire loop is an important aspect of the experimental assembly. The evaporator plate and the cold plate had to be instrumented with thermocouples to ascertain their performance. The pressure drops throughout the system must also be measured and calculated. In addition there are four ammeters in the electrical loop used to measure the input power to the evaporator plate heaters. All of the various instruments and testing devices were combined, monitored, and displayed and controlled by the data acquisition unit. The program that controlled these various sensors was designed using a LabVIEW program specifically written for the CAT loop. The program was also responsible for storing the results of each data run. Each of the various sensors will be discussed before describing how LabVIEW governs their relationship.

5.4.1 THERMOCOUPLE MOUNTING

To measure the various temperatures of different regions of the loop, thermocouples are used. A thermocouple consists of two wires of dissimilar metals that are electrically connected at one end. Based upon the Peltier effect, the temperature of the junction creates a voltage difference that can be related to temperature. There are many different types of thermocouples based upon the different combinations of metals. Type K thermocouples were used for this experiment. A pictorial view of a type K thermocouple is shown in Figure 47.

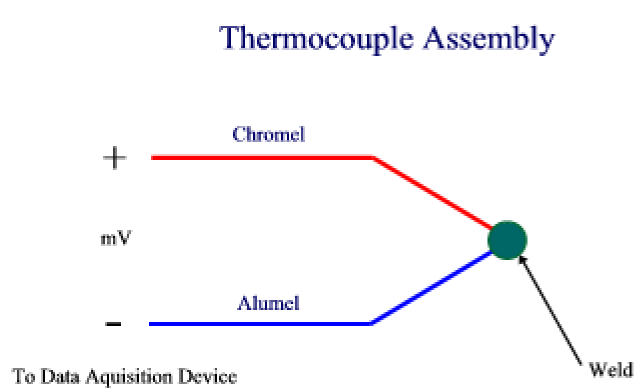


Figure 47: Type K Thermocouple Assembly

There were 68 thermocouples used in measuring the temperatures at various parts of the loop. The three most important regions are the cold plate and both sides of the evaporator plate itself. A side view of the evaporator plate with a thermocouple map is shown in Figure 48.

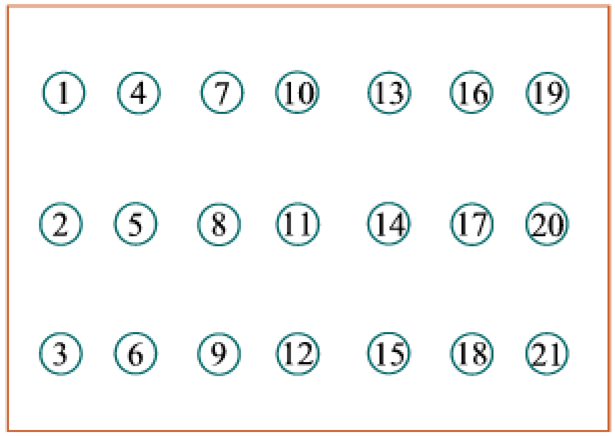


Figure 48: Thermocouple Map of Evaporator Plate

Each side of the evaporator plate has twenty-one thermocouples attached at the above locations. These thermocouples allow for the reconstruction of a temperature contour map to illustrate what is happening during the operation of the plate. The cold plate is the other important region for instrumentation. A schematic of the cold plate with instrumentation is shown in Figure 49. The thermocouples have been renumbered for simplicity.

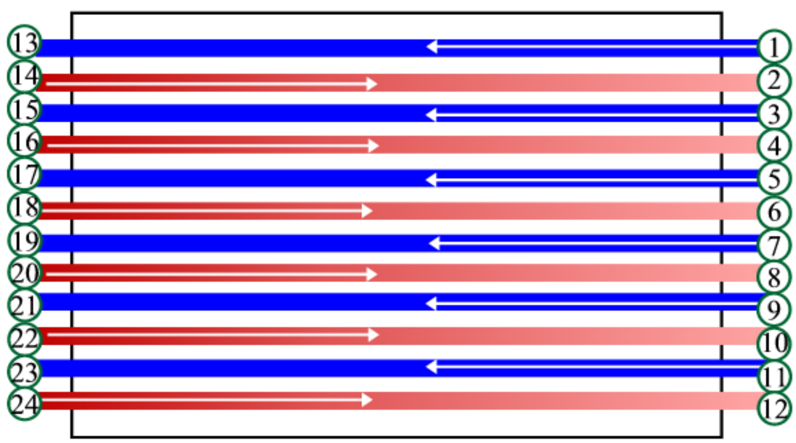


Figure 49: Cold Plate Thermocouple Map

Both the chill water lines and the CPL lines are instrumented to ascertain the temperatures of the inlet and outlet temperatures. This data was then used to determine the amount of subcooling and also quantitatively determine the overall heat transfer coefficient of the cold plate.

5.4.2 WATTMETERS

To measure the power applied to each heaters, ammeters were used to measure the current. A Variac transformer was used to control the AC voltage applied to each heater. As the power in the heaters rises, the current through them also rises. Using the relationship that P=IV, the current can be used to ascertain the power being input to the heaters. The current and voltage were input to a watt transducer which generated a 4-20 mA output which was used to correlate to a power wattage from 0-1000 W for the initial heaters. Since the heater current was stepped down fivefold, the actual heater output could range from 0-2000 W per Variac.

Each of the heaters used was a 900 W electrical resistance heater, which can take up to 208 V AC of applied voltage. Each wattmeter monitors the current in a pair of heaters. The basic heater map of the location of the heaters on the evaporator plate is shown in Figure 50.

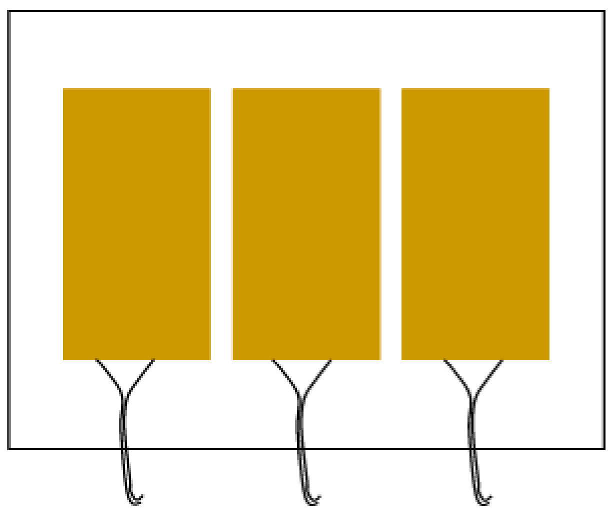


Figure 50: Basic heater arrangement for both sides of the evaporator plate

The heaters were affixed by thermal grease to the copper thermocouple plates on one side and insulated by marine plywood on the other side. This ensured that all of the heat being generated by the heaters went directly into the evaporator plate. As shown in the electrical schematic each Variac controls two heaters. The voltage was controlled by the Variacs, which were used to adjust the heat load applied to the plate.

For the tilt and pitch experiment the 900 W heaters were replaced by 1650 W heaters arranged in the following configuration illustrated by Figure 51.

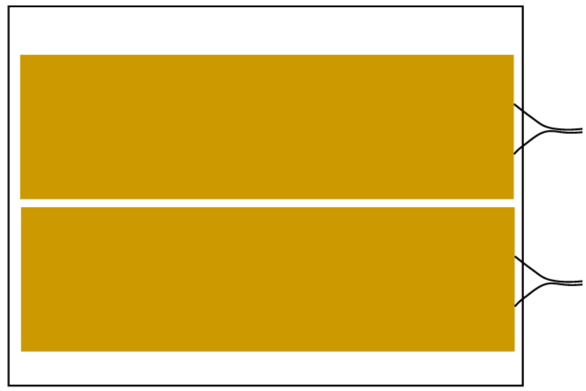


Figure 51: Heater Arrangement During the Tilt and Pitch Experiment

The LabVIEW program was adjusted to reflect the new heaters and the testing proceeded as before. These new heaters were added to replace several burned out 900 W heaters and to better blanket the surface of the copper plate.

5.4.3 PRESSURE SENSORS

There are four pressure sensors in the CAT loop. They are located by the purple color on the CAT schematic, Figure 45. Two of the pressure sensors are differential, meaning that they measure the difference in pressure between two locations. The other two pressure sensors in the system measure the absolute pressure. The absolute pressure sensors have a reference junction that is a sealed vacuum and the pressure at the other junction is compared to the vacuum. The pressure sensors operate under the principle of measuring a voltage difference between the two pressure junctions. This voltage difference is then translated into a signal which is fed into the data acquisition unit and the measured voltage is plotted using the calibration curve and fed onto the display. Instead of having a reference junction at a vacuum, the sensor just compares the voltages from two separate nodes to one another and reports the difference.

One absolute sensor measures the internal absolute pressure of the liquid return line and is used to determine the absolute pressure of the evaporator plate. The other absolute sensor is located on the vacuum pump and is used to monitor the amount of vacuum drawn on the system before operation. Before operation, the pressure sensors were all calibrated at one atmosphere to ensure that they are reading correctly. The first differential pressure sensor measured directly the pressure drop caused by flow through the wick and evaporator plate. The second was used to measure the pressure drop across the needle valve. These data points were used to analyze and study the performance of the plate as it operates in both transient and steady-state modes.

5.5 THERMOCOUPLE PLATES

Two thermocouple plates were designed to mount the thermocouples so that they will be in direct contact with the evaporator plate. The heaters were then mounted on the other side of the thermocouple plate. These plates allowed the heaters to lie completely flat on the surface with no air gaps. If the heaters were placed directly on top of the thermocouples and thermocouple

wire, they would not lie flat and would create air gaps. These air gaps increase the thermal resistance and could cause the heaters to burn out or the thermocouples read the temperature of the air gap. The thermocouple plates eliminate both of these problems. The plates themselves were made of copper because of its high thermal conductivity. The plates do not inhibit the flow of heat from the heaters to the evaporator plates. Lateral conduction in the thermocouple plates will allow some spreading of the heat input across the plate. The thermocouples were placed in a machined groove in the surface of the plate to allow them to lie flat. They are held in place with a thermal glue. A picture of the thermocouple plates before the thermocouples and heaters were added is shown in Figure 52.

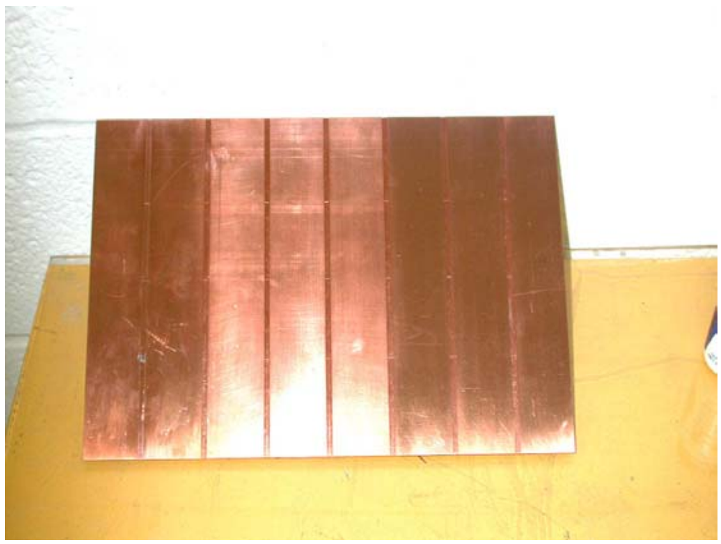


Figure 52: Thermocouple Mounting Plate

A plate was designed for each side of the evaporator. Each plate has the capacity to hold 21 thermocouples. These thermocouples provided a contour plot of temperatures on the evaporator plate surface. The thermocouples will show any hot spots or localized areas of heating.

5.6 DATA ACQUISITION UNIT

The temperature, pressure and current sensors are controlled by a National Instruments SCXI – 1000 Data Acquisition device (DAQ). The data returned from the DAQ is read by LabVIEW virtual instrument panel. The instrument panel is the primary user interface in controlling the system. The panel shows all of the relevant data and also provides a mechanism to save the data in a spreadsheet form so it can be accessed later for a more in depth analysis. A screen shot was captured off of the LabVIEW program and is shown below as Figure 53.

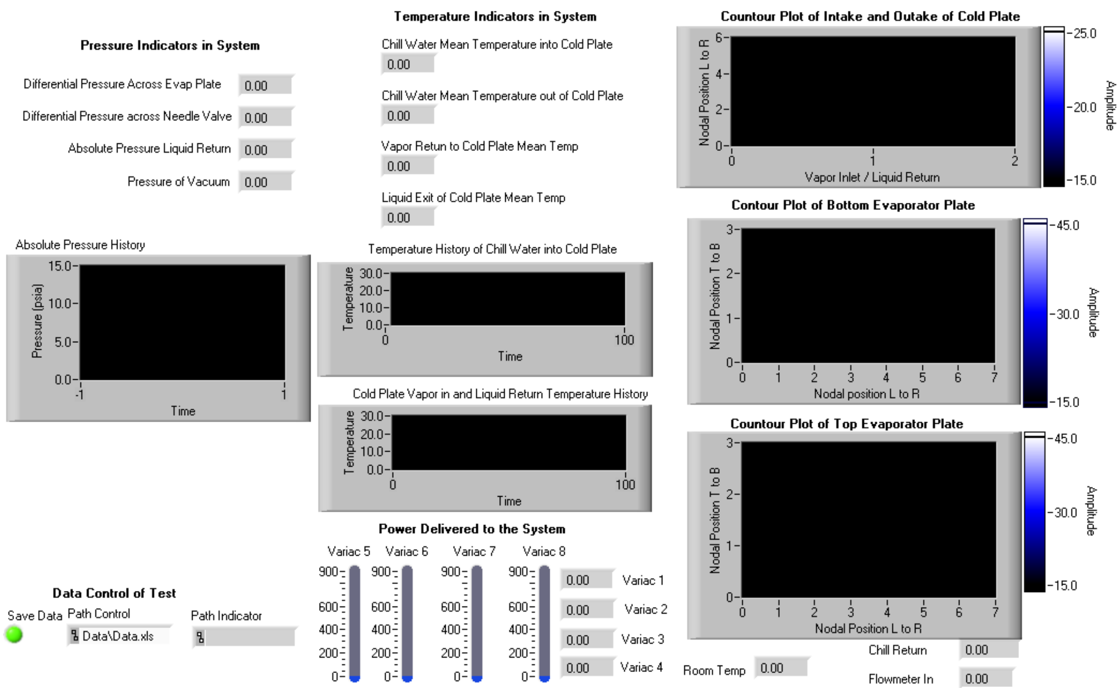


Figure 53: LabVIEW Instrument Panel. This panel allowed the user to control all of the variables in the lab. It also displayed all of the collected data.

In this system, there are three circuit boards that measure voltage and thus can be individually configured to receive signals from a pressure sensor or a thermocouple. The fourth board is a current board and acts as the ammeter in the system. Each one of the pressure sensors, watt transducers, and thermocouples was wired up to the appropriate board individually and tested to ensure its operation.

The data acquisition unit places all of the data it records into a long string where it is analyzed in the LabVIEW program and displayed. A schematic of the LabVIEW virtual instrument programs created are attached and listed in Appendix 2. These schematics show how the data was obtained from the DAQ and modified to show the corrected temperatures, pressures, and power. The schematics also show how the incoming data was organized and sorted into the correct categories and plots. The LabVIEW Instrument Panel and associated schematics were all created specifically for this Trident Project and the CAT loop.

5.7 CONCLUSIONS

The laboratory set up for the testing of the CAT loop is complex. There are four separate physical loops that compose the CAT system, only one of which is the CAT loop itself. The instrumentation and control of it is just as important as the design of the evaporator plate itself. By creating a simple and easy to use program to acquire data during testing, the measured temperatures and pressures were easily interpreted and modified. This section also introduced all

of the independent variables that were measured and recorded during testing including: chill water temperature, chill water flowrate, heat input, tilt, pitch, and the pressure differential across the pressure restriction valve. The next section details the different tests that were conducted using the CAT loop and its associated instrumentation.

6.0 CAT TESTING

Several tests were performed upon the loop to ascertain its performance under varying conditions. Different parameters studied include basic loop operation and start-up, subcooling, tilt and pitch, and performance under pressure restrictions. The startup of the loop became an important aspect of the testing phase and several important conclusions and facts were drawn from the first initial runs of the CPL. The issues and problems with the start-up, priming, and initial operations of the loop are discussed first, followed by a detailed discussion of the different tests that were run on the system.

After the initial testing runs were completed, the subcooling experiment was conducted using the aluminum evaporator plate. After the subcooling runs were completed, the aluminum plate was replaced by the copper evaporator plate for the tilt and pitch experiment and the pressure restriction testing.

6.1 INITIAL TESTING

There were several initial tests that were performed as the course of experimentation began. Before any actual loop operation could take place, the loop itself must hold a vacuum, keeping the air outside from leaking into the system. Thus, the first preliminary test was a vacuum check of the entire system. During this examination, leaks were found in the seal of the aluminum evaporator plate. After the best seal had been established in the evaporator plate, several initial loop startup runs were conducted to ascertain the performance of the loop and the proper startup procedure.

These initial runs were the first attempts to actually operate the loop and at this point there were many unknowns about the performance and heat removal ability of the CAT loop. The first tests varied the amount of working fluid, the configuration and intensity of the heaters, and the flow rate of the chilled water to see their general qualitative effect on the loop. Therefore these first tests were important in establishing the correct amount of working fluid to use in the loop. The initial runs were also run at a variety of different heat inputs with the sink temperature at room temperature to establish a performance baseline. The initial testing was also responsible for discovering the effectiveness of the evaporator plate by observing the spread of temperatures across the surface of the plate. To avoid having the heaters influence the thermal contour plots, the plate was run with heaters only active on one side and observing the temperatures on the reverse side. This test was designed to show how even the temperatures were inside the plate itself.

6.1.1 VACUUM TESTING

One issue with the preliminary testing involved developing a solid vacuum seal on the evaporator plate. The evaporator plate was initially sealed with a silicon gasket material, RTV, around the plate's perimeter. The RTV was placed in the tongue and groove and the two halves of the plate were clamped together until the RTV harden and set up. A picture of the assembled and sealed plate is shown below in Figure 54:

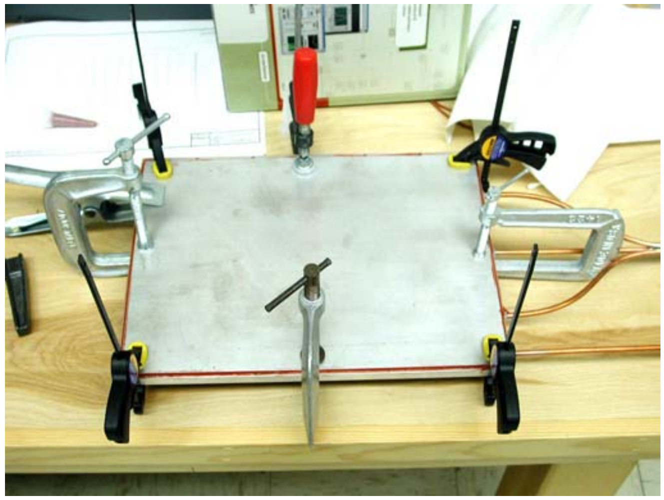


Figure 54: Assembled and sealed aluminum flat plate evaporator

The plate was then installed into the loop and placed under a vacuum using a vacuum pump. The absolute pressure sensors in the system indicated that there was a leak somewhere in the loop. Thus a two step process was developed to check the loop for leaks. The plate itself was checked for leaks by slightly over pressuring the loop with nitrogen and spreading soap around the perimeter of the plate. Leaks were identified by bubbles forming and growing from the escaping nitrogen inside. There were several leaks that developed before the seal blew from the overpressure. With the plate temporarily out of service, the physical lines where checked for leaks by capping off the lines that lead to the plate and a slight overpressure was then added to the lines and again the soap was used to detect a leak. The plate lines had only minor leaks which were eliminated by tightening the connections.

The plate was cleaned up using acetone and steel wool and the process repeated, clamping tighter this time. The plate still had a slow leak in it when the initial battery of tests was begun on the plate in thermosyphon operation.

6.1.2 INITIAL THERMOSYPHON TESTS

The initial thermosyphon tests were run with a 3.5 GPM chill water flowrate at room temperature and an initial heat flux of 166 W over both sides was applied. The initial charge calculation called for a working fluid volume of 300 mL in the loop. This calculation was just based upon an estimation of volume of the available space inside the evaporator plate and the lines connecting the cold plate to the evaporator plate. The loop did not function properly because the volume of working fluid was too large for the available volume. A plot of temperature vs. specific volume for water shows why the loop did not work as planned in Figure 55.

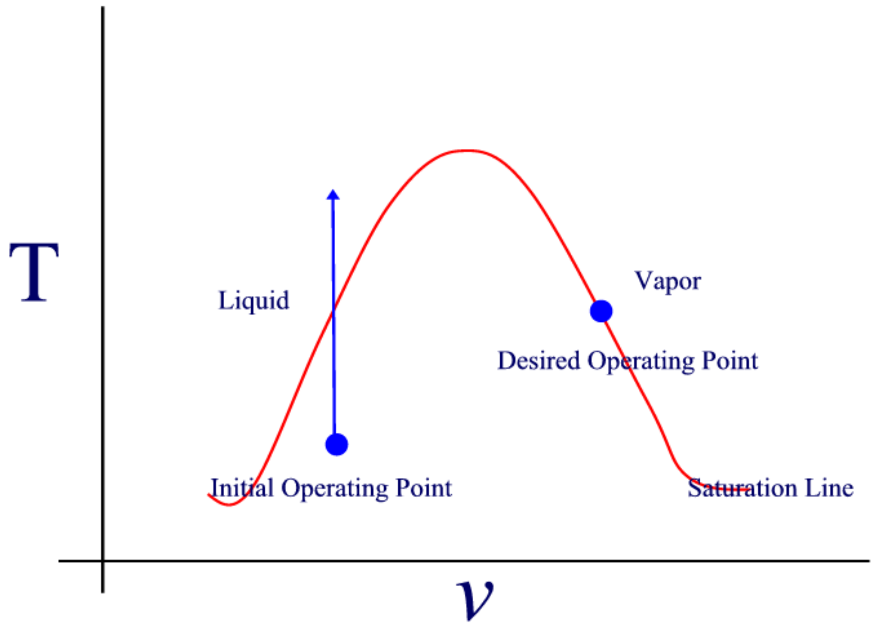


Figure 55: Low Specific Volume Saturation Curve

By decreasing the actual mass of the working fluid, the specific volume can be increased and the operating point can be pushed to the right ensuring that the loop will operate properly. In the initial start-up cases, heating the water up did not start evaporation but served to create a compressed liquid. The desired operating point is on the vapor side of the saturation dome where any increase in temperature would cause the liquid water to evaporate instead of becoming subcooled and compressed.

In addition to adding too much working fluid, the leaks in the evaporator plate became more significant over time, so after the data from the initial testing runs was obtained, the evaporator plate was once again removed from the loop and the RTV removed and reapplied. After the RTV hardened, the plate failed the vacuum test for a third time and the decision was then made to use an epoxy to seal the plate. The epoxy created a permanent seal around the plate but was problematic as air eventually reentered the plate.

6.1.3 CONCLUSIONS DRAWN FROM INITIAL TESTS

Ideally welding or soldering the two pieces together would have obtained a better seal for the aluminum plate. However, the wicks inside the aluminum plate were plastic and would melt at the temperatures needed to solder the two halves together. The RTV seal was seen as the next best alternative because it could be easily removed. Its repeated failure led to other alternatives to be looked at as solutions to the sealing problem. This forced the decision to epoxy the plates together, permanently joining them.

Another problem with the effectiveness of the seal was the thickness of the top plate. To achieve the specified overall dimensions, the top plate had to be no thicker than an eighth of an inch. The thin plate buckled slightly during machining as the mill removed material off of the surface of the plate, causing it to become stressed. The aluminum top plate is not flat anymore but buckled convexly upward because of the material stresses inside the aluminum plate. A thicker plate would not exhibit as much deflection and buckling during the machining process.

Better loop start up and performance was achieved using uneven initial heating along with a small initial system charge. Once the system was primed and operating, additional working fluid could be added to the system during operation. This should prevent the subcooling from occurring during the initial testing runs.

The final lesson learned from the initial tests was that the constant handling of the evaporator plate is also a negative aspect. The intake and outtake lines affixed to the plate were attached and unattached to the rest of the loop's plumbing, repeatedly weakening them and causing failure at the joints. The vapor and liquid lines had to be reaffixed to the plate several times. These were also areas where the plate leaked at one time or another. The plate should be handled minimally to avoid further failures in the joints.

The initial testing data showed that the instrumentation of the loop and the data acquisition system were functioning without any inconsistencies or errors. During the tests, data was recorded every second. In addition, all of the pressure sensors and thermocouples functioned correctly. Once the plate held a vacuum, the major testing phase was begun.

The design of the copper plate was modified before it was placed in the loop to ensure a better seal. The lines into the copper plate were brazed in place to ensure that they would not come loose no matter how much the plate was handled. A half inch of material was also added around the perimeter of the copper plate and bolt holes were drilled. After the wicks were sintered into place and aged, the plates were assembled. The copper plates were sealed by two separate beads of RTV and then held together by bolts around the edges.

6.2 OPERATIONAL NOTES AND TRANSIENT EFFECTS

After running the loop several times there are several important parameters and behaviors exhibited by the loop that have been observed. The aluminum plate was much harder to start up and prime than the copper plate. The differences observed between the two plates during operation and their effect upon the CAT loop will be now be discussed. The procedure for priming the aluminum plate is described below.

6.2.1 ALUMINUM PLATE OBSERVATIONS

The loop functioned best with a liquid fill of 250 mL, added before operation. The start-up procedure takes about a half hour to prime the loop for operation. To start the loop, the rear heaters are brought up to 400 W each, and the liquid return valve is almost completely closed. The vapor pressure is then allowed to build up inside the loop until the vapor starts to flow up to

the cold plate. From this point the liquid return valve is opened completely and the loop began operation.

The loop performed better at higher heat inputs than the lower inputs. Below 750 W, a chugging phenomenon occurred, where the mass flow rate of the working fluid was not enough to ensure a smooth steady flow. The working fluid tended to build up in the cold plate until the pressure rose enough to push the liquid into the evaporator plate. This chugging operation at the low heat inputs was caused by the air leaking into the system. After 1000 W, the loop experienced steady flow and the pressure actually decreased because the higher heat input generated enough of a pressure head to push the air out of the cold plate.. In addition the air leakage pushed the operating temperature of the plate up to 100 C. Even though the loop takes about 30 minutes to start up, it has a very fast transient response. After changing the flow rate or input heat, the loop would reach steady state after about a 3 or 4 minute transient response. This fast response is due to the small thermal mass of the plate that makes it very responsive to the outside inputs. Thermal mass refers to the total thermal capacity of the plate. The actual mass of the plate plays the most important role in determining transient effects because a heavier plate will take longer because it takes more energy to effect a temperature change.

The evaporator plate design was very effective at spreading the heat across its entire area. One of the concerns during the design phase was that of localized cooling, where the only active portion of the evaporator plate would be the left hand side closest to the vapor and liquid lines. However, even at the lower heat fluxes the liquid spread across the top of the plate and dripped down, wetting the plate evenly, as evidenced by the thermocouple measurements.

Both the plastic wick and the heat spreaders functioned exactly as designed while the evaporator remained in the vertical orientation. Both sides of the plate surface maintained nearly constant temperatures during operation.

A baseline temperature profile was taken by turning one side of heaters off and observing the opposite side. This profile was taken because as the plate operated under a partial vacuum, air gaps were created between the evaporator plate and the thermal couple plates. The thermocouples over the airgap measured the temperature of this air gap. This baseline established a pattern that could be analyzed for trends during the pitch and tilt testing. The temperature profile is shown in Figure 56.

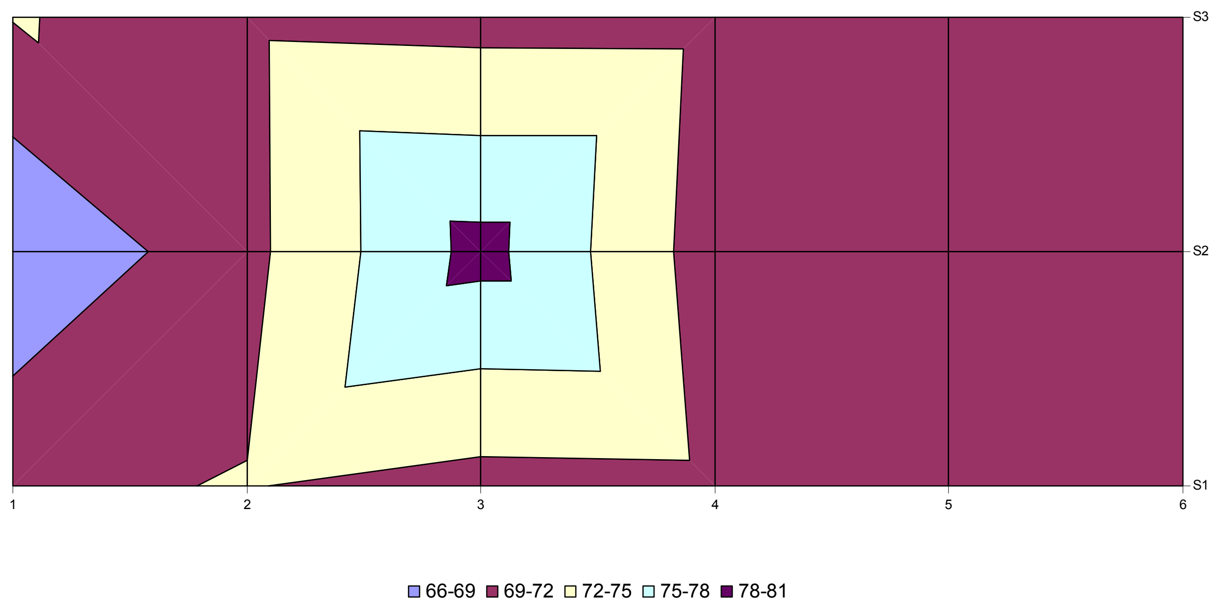


Figure 56: Baseline Temperature Profile of the Top Plate. This profile was created by applying heat from the bottom plate. It was obtained using room temperature chilled water at 1000 W of total heat input.

The contour plot shows the distribution of the temperatures across the surface of the evaporator plate. The resolution on this plot is 3 degrees C. The uniformity of the temperatures is readily apparent from this plot. The small temperature rise in the center is due to a small air gap between the thermocouple plates and the evaporator plate itself. This profile remains approximately the same throughout all of the testing. The only changes are in the magnitude of the observed temperatures. The air gap shows up hotter than the rest on all of the other contour plots as well. This plot will be referenced as the baseline case for the rest of the testing.

Figure 56 shows how the heat spreaders and the wick function to spread the heat evenly over the opposite side. To reach the thermocouples the heat had to travel from the heaters into the evaporator plate and then through the vapor space and heat spreaders to the other side of the plate. This shows the effectiveness of the evaporator plate design at ensuring that the heat is distributed evenly throughout the plate.

6.2.2 COPPER PLATE OPERATIONAL NOTES

The copper plate was much easier to start up and prime than the aluminum plate because of the better vacuum seal which meant that no air leaked into the system during operation. The procedure for the start-up of the copper plate involved first pulling a vacuum on the plate, then turning the heaters on to 400 W each and waiting for the plate temperature to rise to about 100 degrees. The working fluid charge of 250 mL was then added to the loop which immediately became primed and began to function. This start-up process takes about fifteen minutes to engage the loop. While the startup is much faster than the aluminum plate, the copper plate also took much longer to reach steady state because of its larger thermal mass.

The copper plate performed equally well at all of the input heat rates. Intermittent flow is observed below 500 W of total heat but the plate functions over a range of heat inputs from 400 W to 3200 W without degradation in performance. The temperature and the pressure rises with the rate of heat input, but the plate continues to function regardless of the outside conditions. Inside the plate, the pressure observed by the pressure sensors directly corresponds to the saturation temperature of the vapor indicated by the thermocouples. This means that the plate was evaporating right on the saturation line of the water.

During pressure restriction testing, as a needle valve was used to restrict the flow, the pressure inside the evaporator plate rose only a small amount. The valve condition was varied from fully open to 7/8 of the way closed. Even when the valve was barely open, it did not affect the performance of the plate. The copper plate functioned under all of the input pressure restrictions, heat fluxes, and tilt and pitch angles. Many of the problems discovered during the initial testing with the aluminum plate were solved by modifying the copper plate. The pressure restriction testing along with the pitch and tilt results showed that the copper CAT loop is very reliable. Several experiments were undertaken to try and cause the loop to fail by extreme angles of tilt and pitch, pressure restrictions, and both extremely low and high heat inputs. None of these caused the loop to cease to function. The reliability of the CAT bodes very well for naval applications. The loop will function in almost any sea state and piping condition as long as heat is applied. The prototype copper plate is evidence of a powerful new way to cool electronics.

6.3 SUBCOOLING TESTING

The first major set of tests were designed to determine how subcooling affects the performance of the CPL. The subcooling testing involved three independent variables: chill water sink temperature, chill water flowrate, and heat input to the evaporator plate. Heat inputs were varied from 250 W to 1500 W. Four different flowrates were tested: 0.375, 1.0, 2.5, and 3.5 GPM. These flowrates were selected because laminar flow in the cold plate transitions to turbulent at 1 GPM. Therefore, two flowrates have laminar flow and two flowrates experience turbulent flow in the evaporator plate. Finally, three sink temperatures were chosen based upon the water temperatures where warships will be operating. The temperatures used in this study were 4, 23, and 37 degrees Celsius. The subcooling tests were conducted using the aluminum evaporator plate.

Subcooling occurs after vapor condensation has been completed in the cold plate. The working fluid enters the cold plate as a vapor near its saturation temperature that then begins to condense as it flows along inside the cold plate tubes. After the vapor has completed its phase change into a liquid, any further cooling of the liquid is known as subcooling. The initial predictions made prior to the subcooling testing began, held that subcooling would increase with increased chill water flowrate through the evaporator plate. Thus the 3.5 GPM flowrate would subcool the working fluid more than the 0.375 GPM flowrate. Less subcooling was also expected at the higher sink temperatures. An illustration of the working fluid undergoing a phase change is shown below to show the subcooling region in Figure 57.

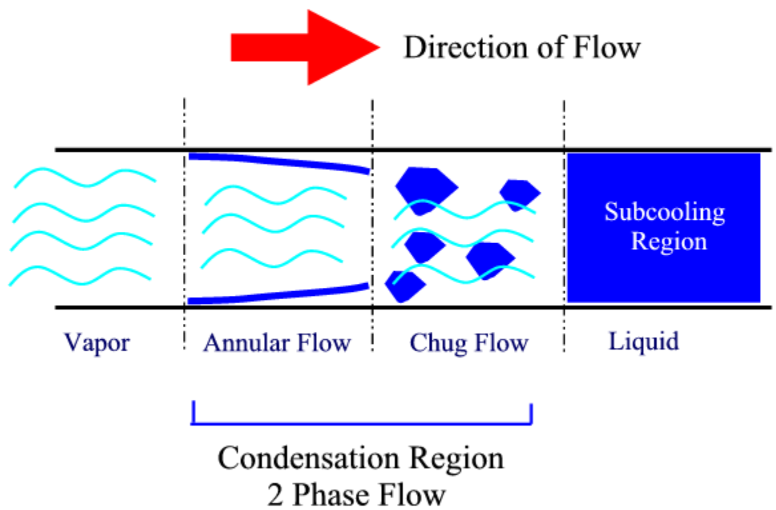


Figure 57: Condensation in flow inside of a tube

Vapor from the evaporator enters the cold plate and starts to condense. Annular condensation begins with liquid forming along the walls of the tubes. As the quantity of liquid continues to build up, the liquid flow detaches from the walls and chug flow results as the liquid droplets are interspersed with the vapor. After the water completely turns into a liquid, the rest of the time it remains in contact with the cooler chill water in the cold plate, the water will be subcooled below its vaporization temperature for a given evaporator plate operating temperature. This brings up the important observation that subcooling, for a given heat input and chill water temperature, is a function of the length of the tubes inside the cold plate.

The first chill water temperature was tested at room temperature, 23 degrees C. The loop was run under the range of heat fluxes and flowrates. The data was all recorded for each setting once the loop reached steady state for each heat flux and flowrate. A total of twelve sets of data were taken. The temperature difference for subcooling was obtained by taking the temperature inside the evaporator plate based upon the saturation pressure and subtracting from it the liquid return temperature out of the cold plate. Subcooling is generally considered to be a negative condition because the liquid returning to the plate must first be sensibly heated to the vaporization temperature before it can undergo the phase change. A plot of the subcooling vs. flowrate at room temperature is shown in Figure 58.

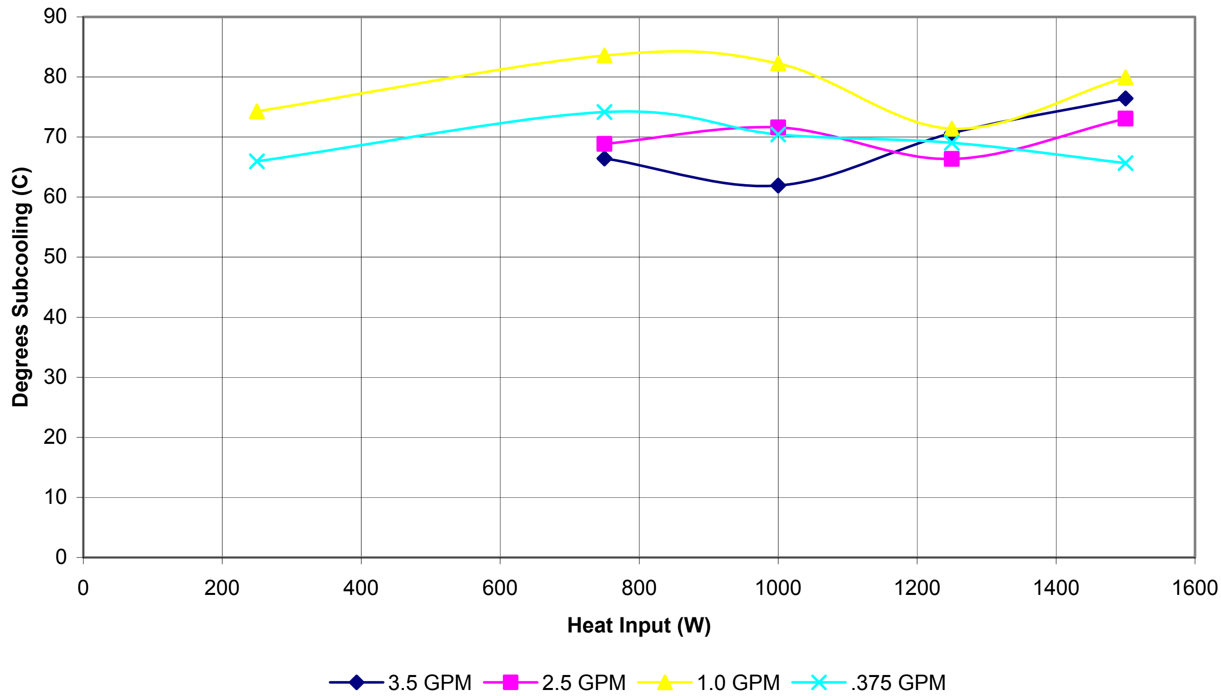


Figure 58: Room Temperature Subcooling (23 C). The data in this plot was taken at all flowrates and heat fluxes for a room temperature chill water reservoir.

The resulting plot shows some very interesting trends. The first conclusion that can be drawn from the plots is the fact that the amount of subcooling is independent, with some allowance for spread, of the chill water flowrate. This is a startling observation because theoretically the subcooling should decrease with flowrate. The data shows that the subcooling does not change significantly with the range of flowrates tested even with the transition from laminar to turbulent flow. The condenser chill water flowrates were chosen so that two of them experienced turbulent flow and the other two exhibit laminar pipe flow.

The next chill water temperature studied was 37 degrees C. It was run using the same liquid fill, heat inputs, and flowrates as the previous experiment. Again, 12 data points were taken of the loop operation at steady state. The results of this experiment is shown in Figure 59:

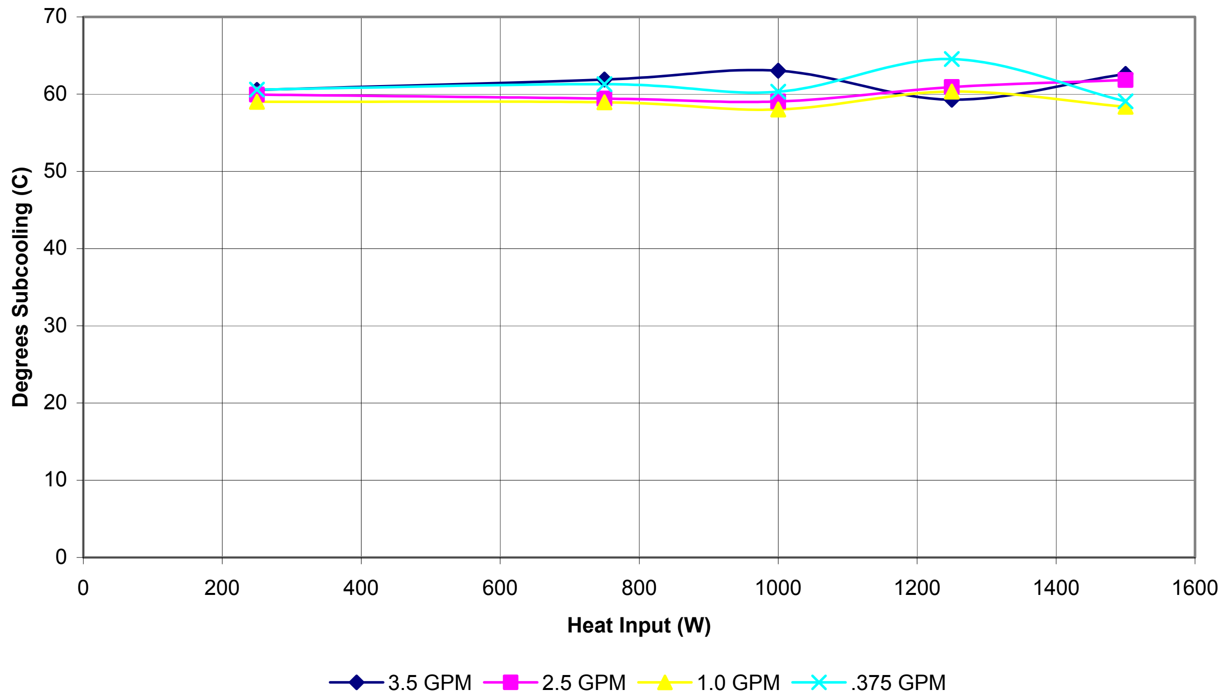


Figure 59: 37 Degree Chill Water Subcooling. The data on this plot was generated from all of the different heat inputs and flowrates.

The data obtained from the 37 degree chill water runs again exhibits the same characteristics as the room temperature chill water. The flow rate of chill water through the condenser does not depend on the amount of subcooling present in the system. It is also important to note that subcooling is not affected by the increase in the amount of heat added to the system.

Finally a sink temperature of 4 degrees was used to simulate a ship in passage in the Artic or North Atlantic. This temperature was tested with the same flowrates and heat inputs as before and the following subcooling results were recorded as shown in Figure 60.

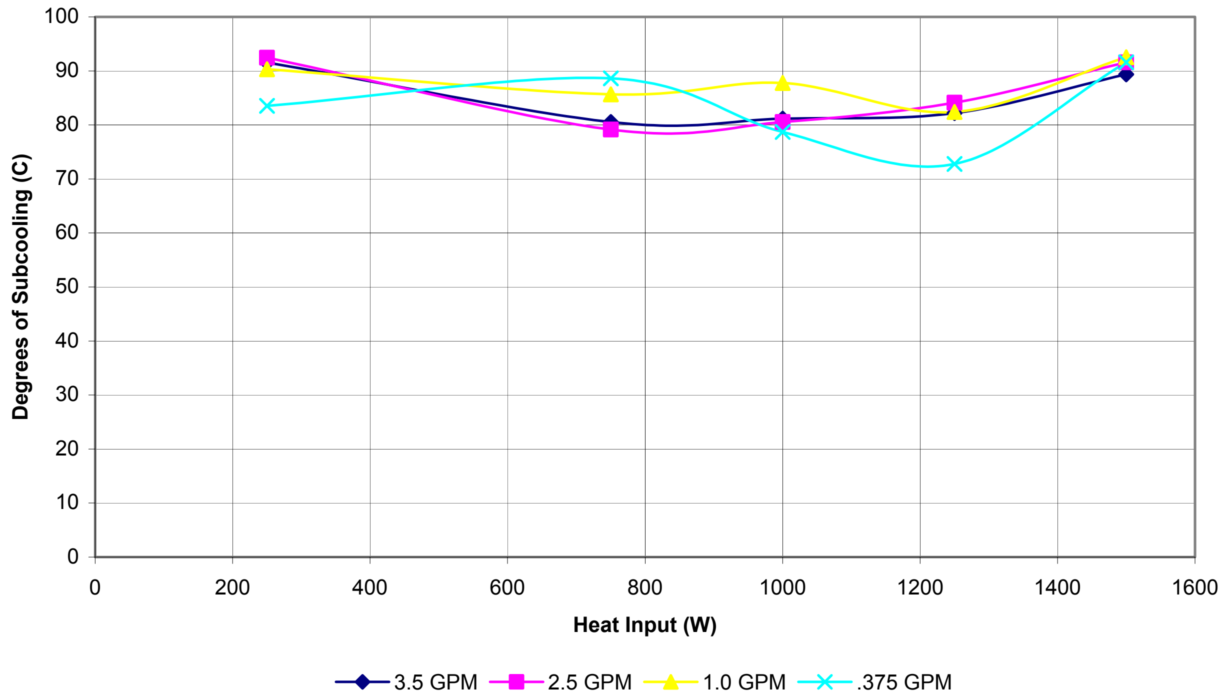


Figure 60: Subcooling observed at a 4 degree chill water bath. This plot contains all of the flowrates and heat inputs taken at the sink temp.

The four degree chill water data again yields a similar result to the previous two chill water temperatures. The flowrate does not have a major effect upon the subcooling in this case as well.

While the subcooling was not influenced by flowrate, there was a strong dependence upon the sink temperature. Figure 61 shows the average subcooling temperature for each sink temperature.

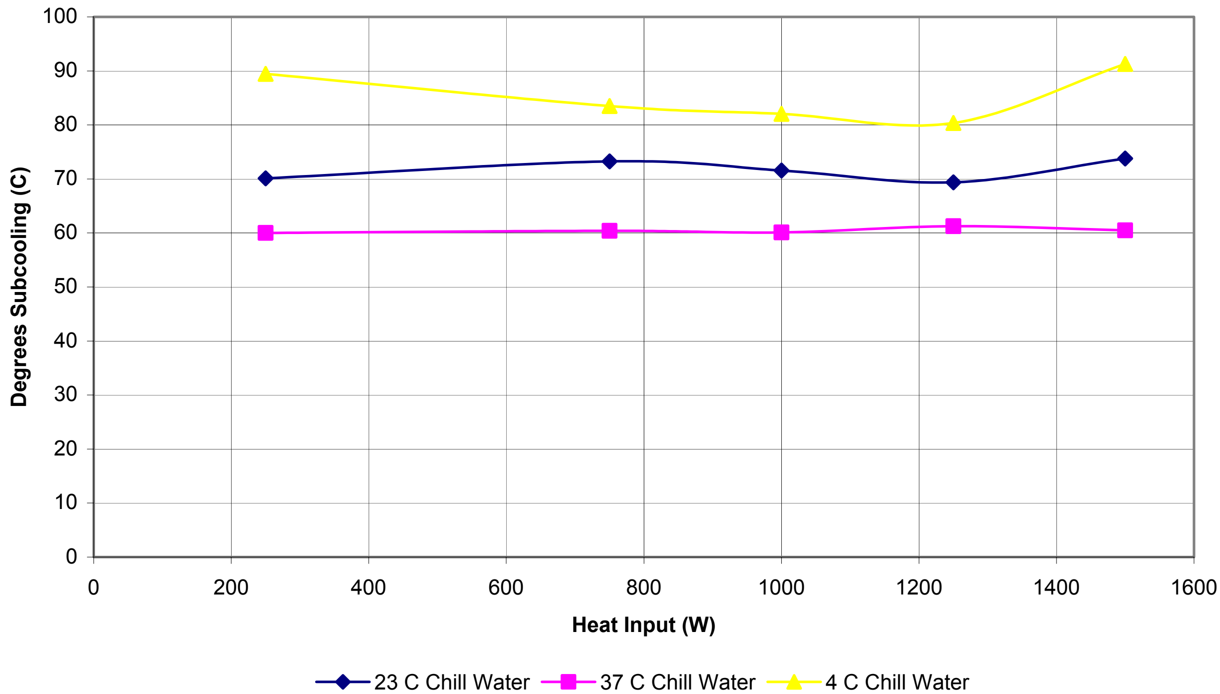


Figure 61: Heat input vs. Subcooling with each data point composed of an average of the amount of subcooling from the four different flowrates.

The major difference between the three plots correlates directly to the temperature of the chill water. Observing a plot like this allows for several important conclusions to be drawn from the subcooling experiment.

6.3.1 SUBCOOLING CONCLUSIONS

The results of the subcooling experiment yielded several important conclusions. The first and most obvious is the fact that both the chill water flowrate and the input heat do not have a major effect on the subcooling observed in the CAT. The temperature of the chill water has the largest effect upon subcooling for the given CAT when air is present.

This also raises the conclusion that the design of the cold plate directly affects the subcooling in the loop as well. A longer condenser means that the working fluid will be subcooled to a greater extent than a shorter condenser. If the cold plate's heat transfer surface is reduced by too great a factor, it will not condense enough vapor and the loop will deprime because all of the working fluid will have evaporated off. The cold plate used in the experimental CPL is too long and will subcool the liquid because of the larger heat transfer region. To reduce the subcooling even further, a smaller cold plate would have to be designed.

A theoretical analysis of subcooling had indicated that a large amount of subcooling adversely affects the performance of the CPL by reducing the mass flow rate. For the three sink temperatures studied during this experiment, the observed subcooling had a minor effect on the

loop's performance because the operating temperatures remained consistent. While this seems contrary to all of the theoretical predictions, a closer look at the temperature data obtained shows that the subcooling did not have a large effect on the mass flow rate of the system. This can be seen by considering the Jakob Number of the CAT loop. The Jakob Number is another dimensionless quantity that relates the amount of subcooling in a fluid system to the latent heat of vaporization. The equation for the Jakob Number is:

$$Ja = \frac{Cp(\Delta T_{sub})}{h_{fg}} \qquad \text{Eq.(20)}$$

The only variable in the above equation that has not been introduced before is the T_{sub}, which is the magnitude of subcooling in degrees. A plot of the observed Jakob Numbers is shown below to determine the effect of the observed subcooling on the Jakob number and the overall magnitude of the calculated Jakob Numbers is shown in Figure 62.

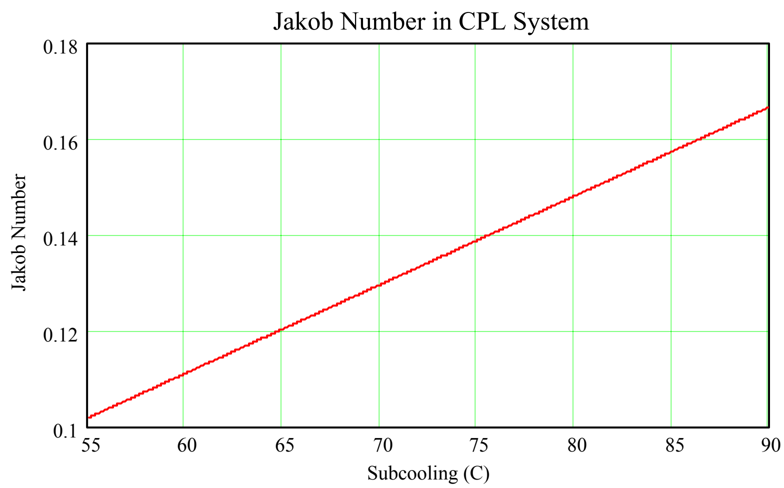


Figure 62: Observed Jakob Numbers during CAT testing. The scale of the Jakob number just shows the range of numbers calculated. A 0 Jakob number would mean that the loop was not functioning.

The Jakob Number varied from approximately 0.1 to 0.17 during CAT operation. This small change has only a minor effect upon the mass flow rate which is shown in Figure 63.

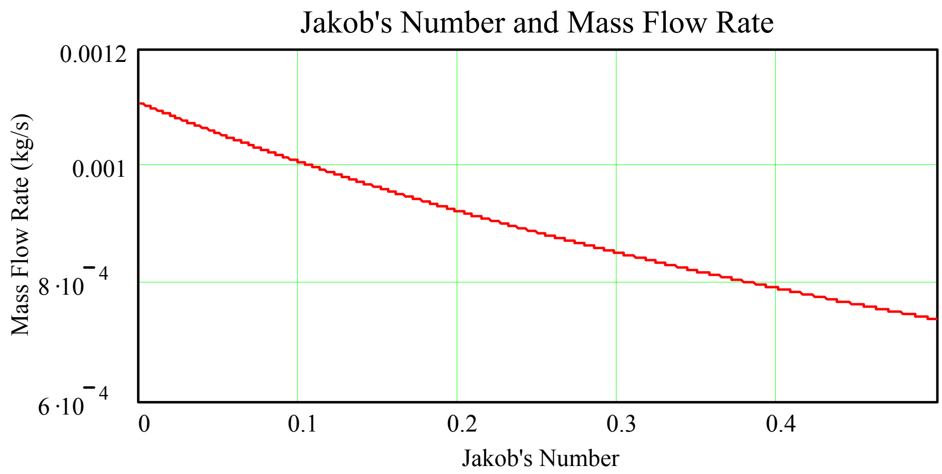


Figure 63: The effect of the Jakob number on the mass flow rate. The region of operation shown by the experimental CAT loop is located between 0.1 and 0.2 on the plot. At higher Jakob numbers the effect is more important

The above plot shows that the Jakob numbers observed during the experiment only vary the mass flow rate by 0.0004 kg/s.

The discussion of the Jakob number and its effects on the mass flow rate of the working fluid in the system predicted that the observed subcooling should only have a minor effect on the operation of the CAT. While the sink temperature affects the magnitude of subcooling, the relative magnitudes of all of the observed subcooling at the different temperatures are small enough to show that the temperature of the chill water does not affect the performance of the CAT. The results also show that the experimental cold plate is oversized and will subcool at any chill water flowrate. For naval applications, as long as the condenser in the CAT is adequate, the performance of a CAT cooling system will not be affected by the temperature of the seawater surrounding the ship. This means that ships employing CATs can travel from the Baltic and the Artic Ocean to the Persian Gulf without noticing any observable effects in the operation of their electronics.

6.4 TILT AND PITCH EXPERIMENT

The pitch and tilt experiments were conducted using the copper plate and wick. The loop underwent startup and priming in the original vertical orientation. Once operational at steady state, the evaporator was pitched or tilted to the desired degree using a digital level which was calibrated before its use to ensure accuracy. The evaporator plate was then tilted and pitched in 15 degree increments from 0 degrees to 45 degrees in the positive and negative directions.

A positive pitch occurs as the front face of the evaporator plate with the liquid and vapor lines rises and a negative pitch occurs as the front end is lowered. The tilt of the evaporator plate was measured from vertical with positive being rotation to the right and negative being rotation to the left. Reference figures number 21 and 22 for a visual representation of this concept.

The pressure inside the evaporator plate was predicted to rise during the adverse pitching, for as the evaporator is leaned forward, the pressure must rise to force the liquid water into the plate. It cannot run into the plate naturally at the adverse pitch. By the same principle, a positive pitch allows a lower internal pressure because as the plate moves below the liquid return line the gravity head increases and does more of the work itself allowing for a lower internal pressure.

The first set of tests was run at 1500 W of heat input, 1 GPM chill water flowrate, and room temperature chill water. The purpose of this test was to ascertain the general effects of pitch and tilt upon the evaporator and loop performance. The heat input was chosen because it results in steady flow. The initial tilt and pitch tests showed that the performance of the CAT loop was only slightly affected by the pitch of the evaporator plate and was almost independent of the tilt.

After verifying the performance of the loop under pitch and tilt conditions, a range of heat inputs was examined. The independent variables included the heat input that took values of 400, 800, 1500, and 2000 W, and the pitch and tilt that were initially tested at 15 degree increments up to 45 degrees.

6.4.1 TILT ONLY TESTING

The tilt of the evaporator plate had no effect on the temperatures of the evaporator plate. The temperature profile of the plate was approximately the same at both extremes. A contour plot of the temperatures is shown below. The first contour plot shows the plate temperatures during a perfectly vertical orientation. The second profile occurs at negative 45 degrees and the third at a positive 45 degrees. Figures 64,65, and 66 respectively.

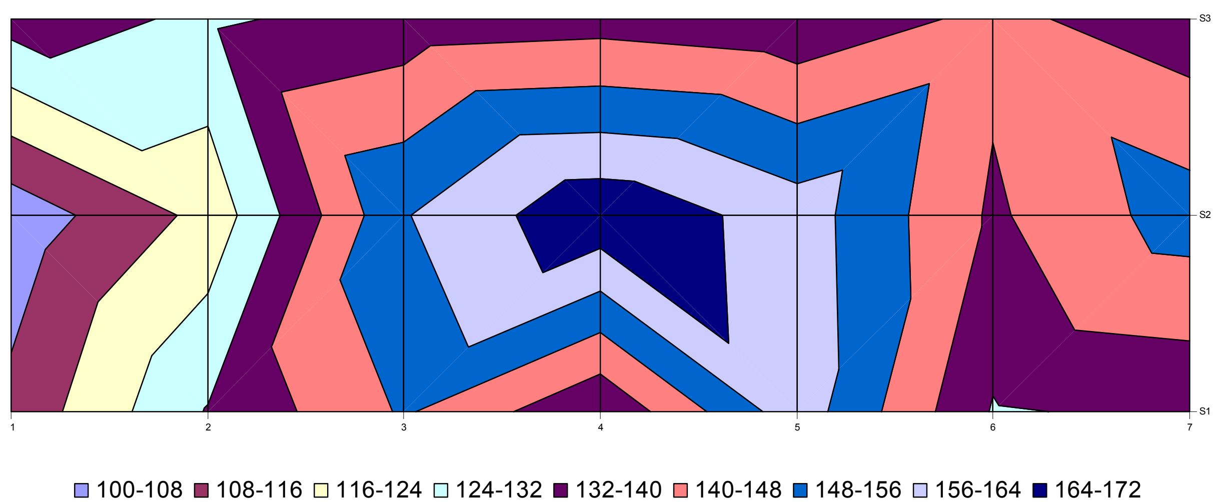


Figure 64: 0-0 Temperature Contour Plot of Bottom Plate with heaters active at 0 degree pitch and tilt.

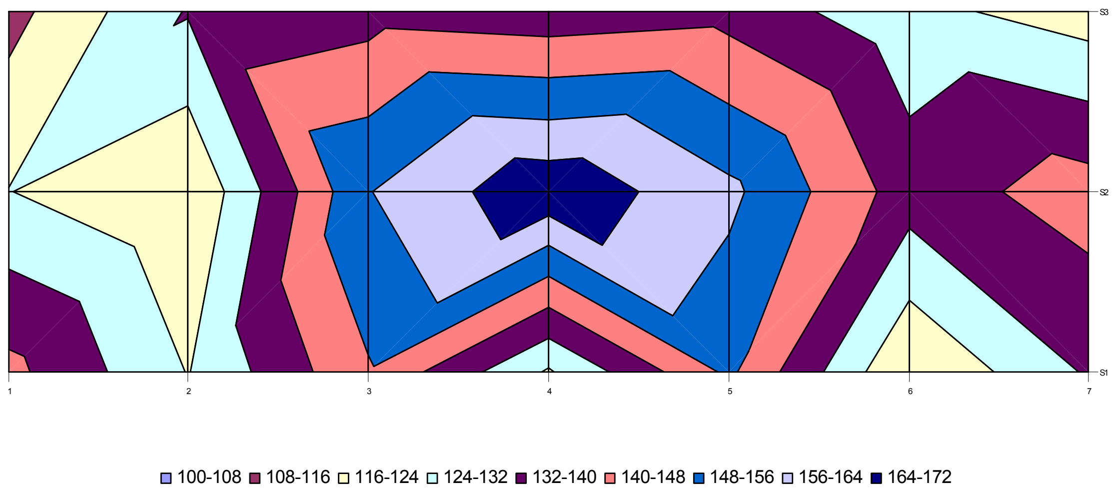


Figure 65: Negative 45 Degree Tilt Temperature Contour Plot

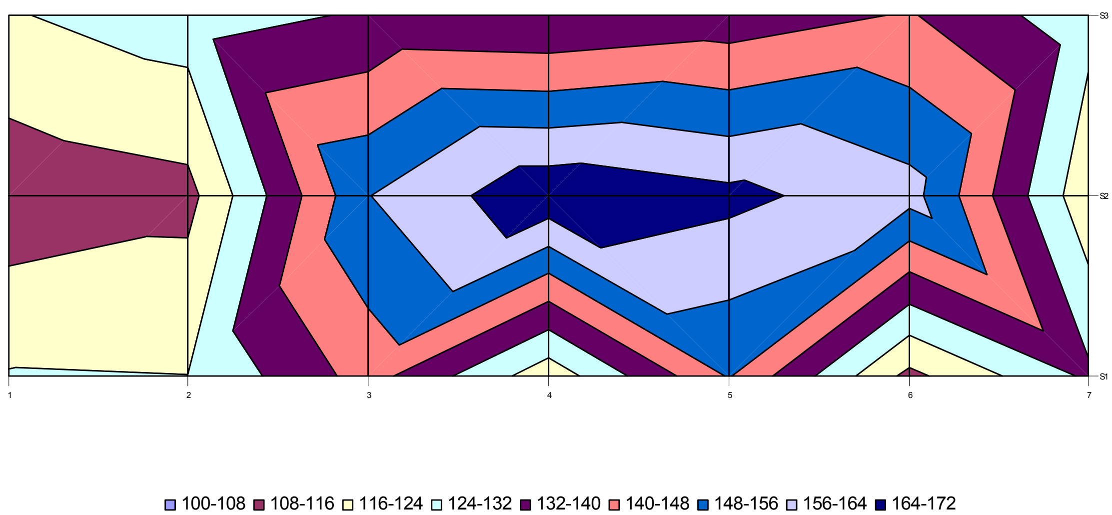


Figure 66: Positive 45 Degree Tilt Temperature Contour Plot.

As indicated by the three previous figures, the tilt did not seem to affect the performance of the loop or the evaporator plate. The plate maintained uniform temperature distribution from the -45 degree tilt to the +45 degree tilt. The hot spot in the middle is a result of the small air gap between the thermocouple plate and the evaporator plate. To see a profile where the effect of the air gap is minimized, turn to Figure 56. The air gap causes a large increase in resistance to conduction heat transfer that in turn causes higher temperatures. The temperature contours only vary a little in their location. These plots show the effectiveness of the copper wick at spreading the water out throughout the working surfaces of the plate. At the –45 degree tilt the bottom plate is lower than the top plate and thus should receive more of the water but it has the same profile as a positive 45 degree tilt where the bottom plate is higher than the top plate. The wick ensures that the surfaces are evenly wetted. If one of the spots began to dry out or flood, the

temperature profile would either sink or rise dramatically. As shown above, this does not happen.

Another way to evaluate the performance of the evaporator plate is to take a look at the internal operating temperature of the evaporator plate and how it is affected by the tilt. Shown in Figure 67 is a plot of how the average operating temperatures in the plate are affected by the tilt.

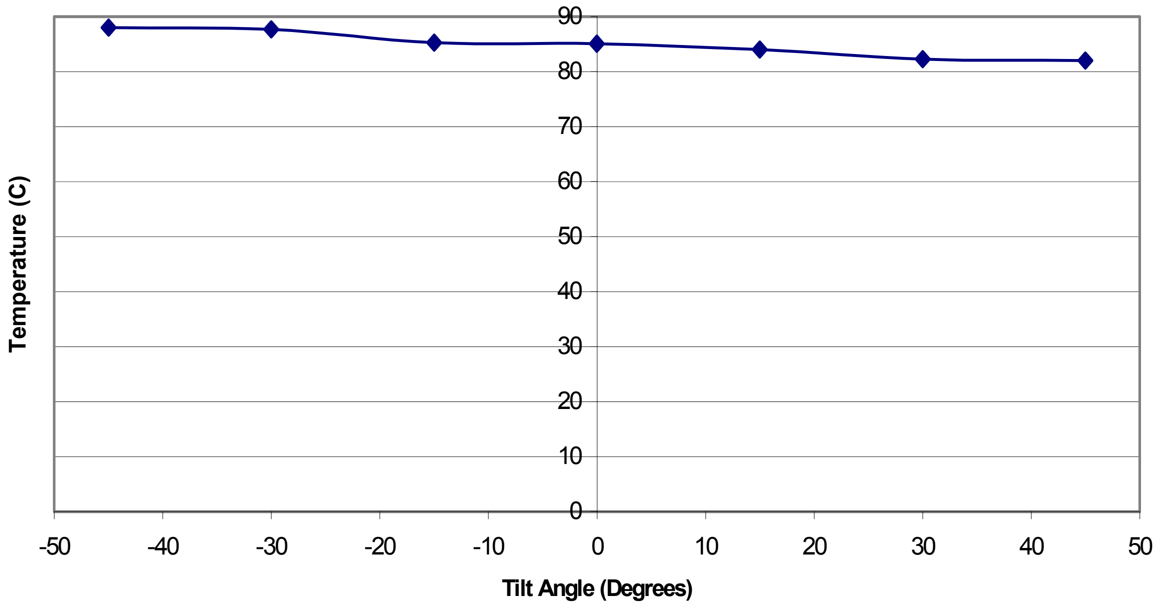


Figure 67: The Effect of Tilt Angle upon Plate Temperature and Hot Spots

The operating temperature only varies by 5 degrees C during operation. Thus at a heat input of 1500 W, the performance of the evaporator plate is not affected by the tilt. The heat spreaders and copper wick are functioning exactly as designed as shown by the even temperatures.

After completing the tilt only experiment, the evaporator plate was then returned to a vertical orientation and the pitch only experiment was begun to see how the pitch affected the evaporator plate.

6.4.2 PITCH ONLY TESTING

While the effect of the tilt upon the evaporator plate and CAT loop was uncertain going into the testing, the pitch of the evaporator was a different case. A positive pitch moves the liquid return line higher than the rest of the plate and aids the flow of the water into the plate as well as making it easier for the vapor to exit as well. This should reduce the operating pressure. When the plate is given a negative pitch, the liquid return line drops lower than the top of the plate. This forces the liquid water to return to the plate upon an uphill slant which should cause the

pressure to increase. Because the pressure and temperature are directly related a higher pressure should yield a higher operating temperature.

The pitch testing confirmed these initial predictions. At 1500 W the evaporator sees about a 20 kPa pressure drop in the positive pitch direction and about a 2 kPa pressure rise in operating pressure in the negative orientation. The operating temperature is highest for the most extreme negative pitch and lowest for the most favorable pitch. A plot of operating temperature vs. the pitch of the plate is shown in figure 68.

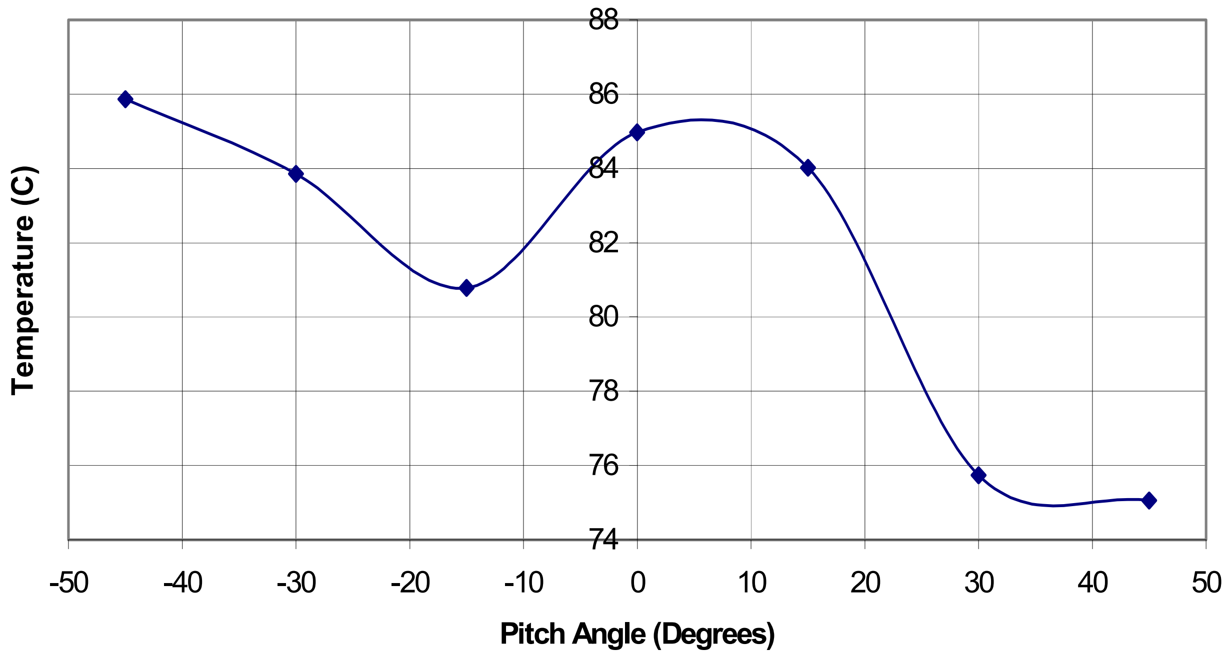


Figure 68: The Effect of Pitch Upon Operating Temperature

Plotting the operating temperature over the range of angles yields a very interesting plot. The highest and lowest temperatures validate the predictions of the theory. However, from the temperature analysis it shows that a small degree of negative pitch makes the water flow easier than a vertical orientation. This small degree of negative pitch removed some of the traps and flow restrictions that the tubing and connections experienced in the upright configuration. The removal of these restrictions would allow the operating pressure and temperature to drop slightly over a level orientation. It is also important to note that the operating temperature only changes by about ten degrees during the whole process at any of the angles involved. Responsible for this small difference are the gravity and capillary pumping head. The small temperature rise is indicative of a slight pressure rise, but the flow around the loop is not stopped.

6.4.3 TILT AND PITCH TESTING

The next step involved with the pitch and tilt testing was to test the extremes of the operation of the plate, determining the heat input and angle where the CAT loop ceased to operate. Ranges of heat inputs were tested from 400 W to 2000 W. This tested both extremes of a high mass flowrate and a low mass flowrate in the most adverse orientation. The evaporator plate was shifted to a negative 45-degree pitch and positive 45 degree tilt and subjected to the various input heat amounts.

The performance of the CAT loop was only slightly affected by the tilt and pitch at all of the input heat loads and angles. The evaporator plate with the circular heat spreaders and the wicks seemed to function exactly as designed during the testing. The semicircular bosses (Figure 35) spread the heat from the higher plate out to the lower wetted surface to maintain a constant operating temperature and pressure just as effectively at the low heat input rates as the higher heat input rates. There is a pressure rise and a corresponding temperature rise in the most adverse conditions of a combination of -45 degree pitch and tilt for all cases but it does not prevent satisfactory operation of the loop. This means that the CAT loop will function under al conditions expected inside a ship at sea. Shown below is a plot of the operating temperature of the plate in the most adverse conditions compared with the operating temperature at 0 pitch and tilt in Figure 69.

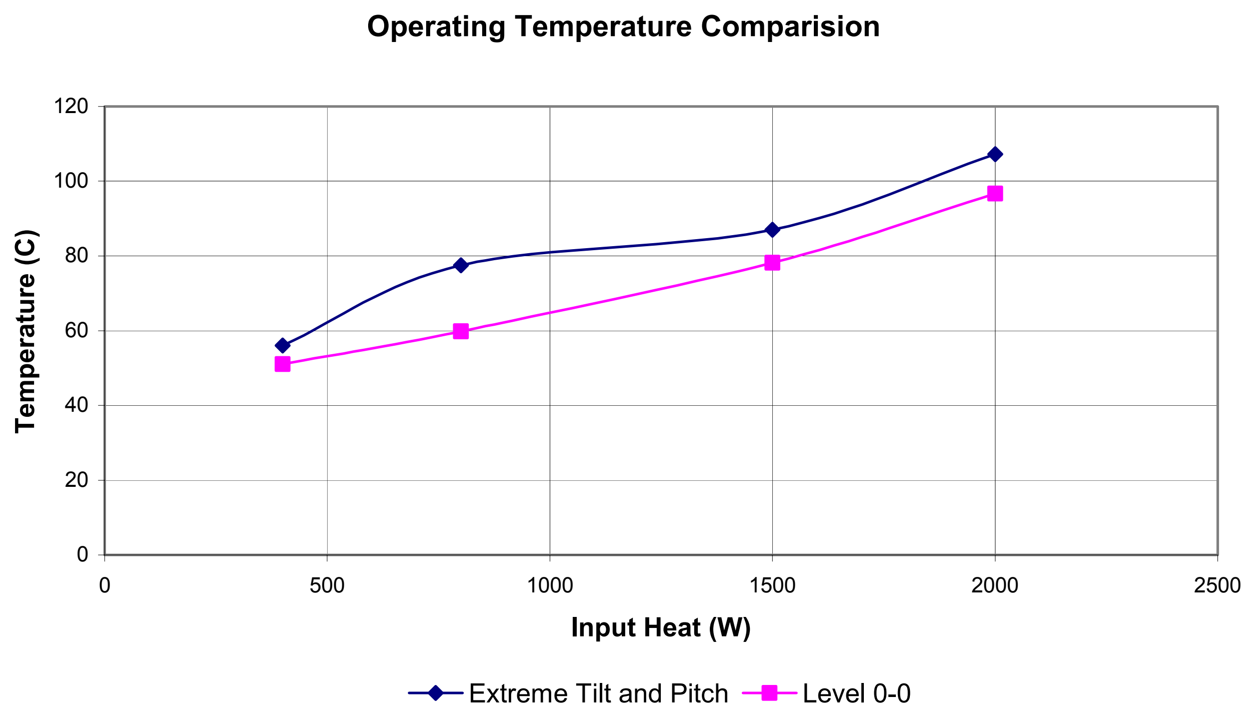


Figure 69: Results of Extreme Pitch and Tilt Testing. Tests took place at 23 C chill water flowrate at 1 GPM.

The temperature in the adverse cases is about 10 degrees warmer than the level cases. This consistency is attributed to the ability of the evaporator plate design to function under all of the input conditions.

6.4.4 TILT AND PITCH CONCLUSIONS

The CAT loop functioned extremely well under all of the input conditions. The loop was able to function consistently at low heat values just as well for the higher heat inputs even under the most adverse circumstances. The combination of the capillary pumping head and gravity head are very effective at ensuring that the flow is maintained despite the difficulties imposed upon the loop. In addition, it took almost ten minutes or more for the loop to reach a new steady state after the change in pitch. This is due to the increased thermal mass involved with the copper plate which has the effect of dampening out the transient effects of temporary changes in tilt and pitch. This means that even larger amounts of tilt and pitch could be withstood by the loop. The effectiveness and consistency means that the loop is well suited to use at sea by Naval vessels.

6.5 PRESSURE RESTRICTION TESTING

One of the measures of the effectiveness of a cooling system is how it overcomes increased pressure drops caused by flow losses. These flow losses can come from obstructions in the piping or simply distances between the evaporator and condenser. To simulate these pressure drops, a needle valve was used to increase the losses in the liquid water as it returned to the evaporator plate. Two different heat inputs were used during this experiment, 500 and 1000 W. These heat inputs were used because the 500 W conditions involves some chug flow and the 1000 W input almost ensures continuous flow. The valve was slowly closed while recording the pressure increase in the entire system. The following plot shows the effect of closing the valve upon operating pressure and temperature in the 1000 W case in Figure 70.

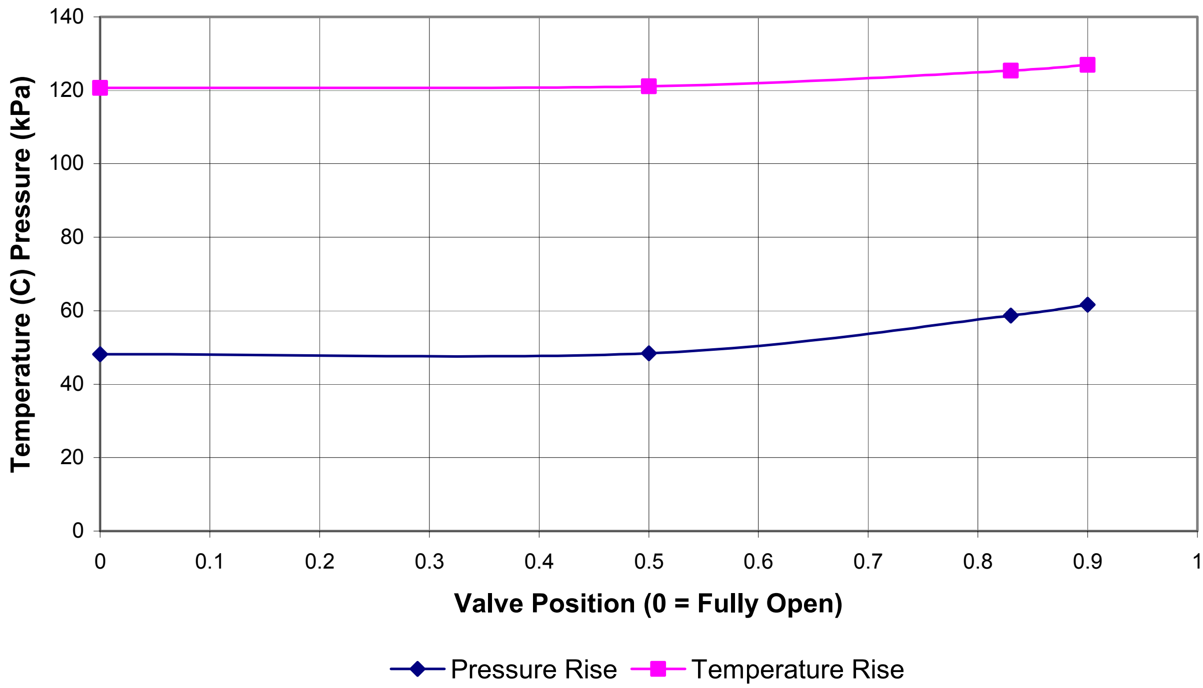


Figure 70: Results of Pressure Restriction Testing performed at 1000 W. Pressure and Temperature have different units but they are placed on the same axis for simplicity in viewing.

As the valve closes, the losses increase necessitating a corresponding increase in operating pressure. However the measured pressure increase was less than 10 kPa. Even when the valve was barely open the plate functioned normally with a slightly perceptible temperature and pressure rise. The only way to stop the flow was to completely shut the valve, preventing the water from flowing. The 500 W case experienced identical results as the closed valve prevented the liquid from returning to the evaporator plate.

Combined with the pitch and tilt conclusions, the pressure restriction testing highlights the benefit of a combination of gravity and capillary pumping heads. Together they produce enough of a pressure head to allow operation despite severe obstructions added into the system. Loop operation was also independent over the range of heat inputs because it works well over the entire range of inputs studied. The CAT loop has turned out to be a very consistent and robust system.

6.6 TESTING CONCLUSIONS

Three different tests have defined many different aspects pertaining to the operation of a CAT loop. Subcooling was found to be independent of flowrate over the range of inputs studied even when subjected to air infiltration. The sink temperature had a much stronger effect upon subcooling than the chill water flowrate. Another conclusion drawn from the results of the testing was the fact that the length of the condenser has a large effect upon the subcooling. Different condensers and cold plates could theoretically be tailored to remove a set amount of heat with minimum subcooling.

The effects of tilt and pitch upon the operation of the loop was the next parameter studied. Despite experiencing angles of 45 degrees on two separate planes, the performance of the loop was not hindered. The flexibility of the loop in handling the different orientations points out the effectiveness of the evaporator plate design. The capillary pumping head in the wick spreads the water out over the entire working surface no matter what the orientation is and the circular heat spreaders transfer heat from the hotter side to the cooler side to ensure even and consistent temperatures.

The pressure restriction testing reconfirmed the effectiveness of the capillary pumping head and gravity head observed during the pitch and tilt testing. Even when the valve was over 90% closed, it only had a minor effect upon the flow in the loop. This test further proved the flexibility of the CAT loop in reorienting and adjusting itself to different imposed conditions.

Using two different evaporator plates during testing also allowed several comparisons to be drawn between the two plates. The aluminum plate, because of its bad seal became air infiltrated. While that meant higher operating temperatures, the loop still functioned and heat was removed from the heaters. The aluminum plate had a lot less thermal mass and thus a faster transient response time due to the lower density of the aluminum. The results of the aluminum plate testing are important because it shows that the loop will still function in the event of a vacuum seal failure. The heavier copper plate had a much better seal and a higher capillary pumping head. The lower operating pressures and temperatures shown by the data taken during the experiments evidenced the greater effectiveness shown by the copper plate. Both plates were very effective and lend their use to naval applications and the validity of the vertical flat plate evaporator as a means to cool electronics.

7.0 CONCLUSIONS AND RECOMMENDATIONS

The capillary assisted thermosyphon loop is a very effective device for cooling electronics because it can handle a wide range of heat loads, 400 W – 3000 W, under conditions that a ship will experience at sea. Even at the most extreme combinations of tilt and pitch, the loop continued to function without experiencing runaway temperatures or pressures. The temperature of the chill water was found to have an effect on the subcooling but almost a negligible effect on the conditions in the evaporator plate. The flowrate of the chill water in the condenser did not affect the performance of the loop with respect to the internal temperatures and subcooling. The only experimental way to cause the loop to fail was to completely shut valves in the system. This translates directly to a reliable device that can serve the Navy's needs in any climate and sea condition.

7.1 EVALUATION OF THE VERTICAL FLAT PLATE EVAPORATOR

There were two flat plate evaporators constructed and tested during the course of this project. The aluminum evaporator plate fitted with a plastic wick was used for the initial testing and the subcooling study. The initial tests identified the critical issue of sealing the aluminum plate to hold a vacuum. The copper plate was modified while it was under construction to add boltholes around the perimeter to improve the seal and thus that issue was successfully resolved. The copper plate also had brazed liquid and vapor lines leading out of the plate to prevent leakage from these key joints. The copper plate was used for the tilt and pitch study and the pressure restriction study. The comparison between the two plates performances yields several interesting conclusions.

The aluminum plate had a much faster transient response time because of its lower mass. It responded much faster to changes to the heat inputs or temperature changes in the chilled water. For naval applications, surges in power are frequently expected and the aluminum plate would be able to deal with the changes easier than the heavier copper plate. The problem with the design of the aluminum plate was in the vacuum seal. The two plates composing the flat plate evaporator are susceptible to deformation during machining because of their thickness. The thin plates tend to bend as stress is relieved from the mill during the machining process. This slight buckling made it harder for the two plates to fit together. The RTV seal by itself was not enough to hold the plate together and prevent air from leaking in. After the failure with the RTV, epoxy was used. The epoxy was initially successful at sealing the aluminum plate but the combination of the high temperature and pressures weakened the adhesive and the epoxy began peeling off the aluminum it was supposed to seal.

Even with the air infiltration, the plate was very effective at cooling the range of heat inputs tested. The air infiltration leads to slightly higher operating pressures and overall temperatures in the plate. On average, the aluminum plate was about 5 psia greater in operating pressure and 20 degrees C higher in operating temperature than the copper plate. The air infiltrated evaporator and loop worked despite the air leakage in, which shows that if the CAT loop was employed at sea and air began to leak into a loop, it would not cause the loop to stop working.

The air entering the loop would not affect the operation of the electronics. This means that the CAT loop does not need constant maintenance or inspection to ensure that it functions correctly.

The copper plate was modified during its construction and assembly to reflect the problems with the aluminum plate. Two bands of RTV were used to prevent the air from leaking in. The RTV bands were allowed to cure before the bolts around the perimeter of the plate were tightened. This ensured that the RTV would not be forced out of the joint between the two plates. The copper wick was sintered into place at Bechtel-Bettis Inc. in Pittsburgh and returned to the Naval Academy for testing. Because the copper plate was able to hold a vacuum it experienced lower operating temperatures and pressures for a given heat input than the aluminum plate. The copper plate also experienced a much faster loop start-up and priming because the water vapor did not have to fight the air leaking into the system. The copper plate was much more effective at the lower input heat rates because it could operate at a lower pressure which meant that the mass flow rate could be correspondingly lower.

The drawbacks to the copper plate include the much higher mass of the copper plate that makes it very slow to respond to transient heat inputs or changes in tilt and pitch. The loop needs much more time to reach steady state. While the thermal mass increases response time, the steady state results from the copper plate are much more repeatable and continuous. Depending upon how much air leaked into the aluminum plate and loop, the pressures and temperatures would change. For the copper plate, a set amount of working fluid and a set heat input reproduces the pressures and temperatures each time. The longer response time benefits electronic devices that generate a set amount of heat and are not impulse driven. The thermal mass of the copper plate will dampen any small transients and will keep the electronics cool under any orientation that the ship will face. However, the higher mass also means a weight penalty as these plates are installed to replace fans. The copper and aluminum plate are almost the same size but there is a large difference in the mass of the two plates.

One final comparison can be made between the two plates is an evaluation of their effectiveness over time. It has already been stated that the aluminum plate lost its vacuum seal with time. The plastic wick inside the aluminum plate is also a source of failure. High temperatures, above 200 C, can melt the plastic and reduce its wicking properties. The wicks inside the aluminum plate were fused together when the plate was taken apart and examined. The wicks had also detached from the walls of the plate and were located in the middle of the vapor space. The life expectancy of these plastic wicks is considerably less than the combination of the copper wick and plate. The corrosion of the copper wick is an issue that needs to be considered and investigated.

7.1.1 DESIGN IMPROVEMENTS

The internal geometry of the plate was very successful at responding to all of the different tests. The semicircular heat spreaders allowed the heat to flow from the hot side to the cooler side and ensured that the temperatures would be evenly spread across the plate. The semicircular heat spreaders also helped to distribute the water over the surface of both wicks. The wicks themselves pulled the liquid water over their surface and ensured that the interior surface of the

plate was wetted allowing for even heat removal across the entire surface of the plate. The plate thickness was held to a ½ inch, which met specifications for existing electronics cabinets. By modifying the design of the plate to include boltholes for sealing, any buckling or strain relief created during the manufacturing process is minimized.

There are several improvements that can be made to the design of the evaporator plate to continue to improve its performance. All of the internal geometry can be maintained because it is very effective at spreading and removing the heat. The design can be better tailored to the desired heat loads by increasing the number of ports into and out of the evaporator plate. The small diameter of the liquid return line limits the flow of liquid into the plate, as do the three vapor lines out of the plate. While the diameter of the lines is constrained by the overall thickness of the plate, a series of small diameters can be formed together in a manifold to facilitate higher mass flow rates for higher heat inputs. The choked flow restriction is the limit to the upper level of the evaporator plate's effectiveness and this limit can be raised by increasing the diameter or number of lines out of the plate which will drop the velocity of the vapor. For the current design, the selection of the three vapor lines and one liquid return line was an arbitrary decision.

Another variable is the number and size of the internal semicircular heat spreaders. Currently there are ten 1-inch diameter bosses in the bottom plate. Their diameter, overall number and placement can be varied to improve the flow inside the evaporator plate. However, the current plate design has proven its effectiveness at distributing the heat under all ranges of orientation and input heat loads.

One of the best areas for improvement would be in the overall size of the evaporator plate itself. The current plate has a surface for heat transfer of 9 by 12 inches on both sides. A smaller plate could be designed, tailored for a smaller cabinet or device. The future of electronics cooling is moving into the effects of very small-scale devices and nanotechnology. Ideally, a CAT loop could be created on the electronics themselves or upon each individual computer chip, processor or switch. The concept of a macro-scale flat plate has been proven and the next step for the designer is to begin to reduce the scale.

Finally, it would be interesting to find a metal wick for the aluminum evaporator plate and to compare the performance of the aluminum-metal wick combination with the copper evaporator plate. The aluminum would allow for savings in cost and weight because of its light weight and low cost compared to copper.

7.2 AREAS FOR FUTURE STUDY

There are many areas for future study that could follow from this Trident Project. There is room for investigation of the effects of cold plate length, and how the length affects subcooling. Another area of study is to directly determine the effect of the wick in the CAT loop by constructing another evaporator plate and assembling it without a wick inside. The plate can then undergo the tilt and pitch experiment and quantify the influence of the wick on the operating temperature and temperature profile across the surface of the evaporator plate.

The current loop could also be configured to run high heat flux devices and see how a localized high heat flux would affect the performance of the plate. Currently, the plate is blanketed by heaters which gives a uniform heat flux rather than a local heat flux. Further studies could also continue to understand the effects of changing gravity head on thermal performance for the capillary assisted thermosyphon.

To further the aims of electronics cooling and its application to the Navy, several evaporators could be constructed and placed in an actual electronics cabinet as a proof of concept. The evaporators would be connected in parallel to one condenser and run simultaneously. The temperature profile of the entire cabinet could then be observed as well as the individual plates.

7.3 PROJECT SUMMARY

This Trident Project accomplished all of the design and testing goals save the 10 W/cm^2 of heat removal. As the local heat flux increased, the plate was able to respond appropriately. However, the heaters were insulated using plywood that tends to burn at high temperatures. The plate was not able to reach the intended heat flux because of safety concerns. The plate was able to cool heat fluxes approaching 5 and 6 W/cm^2 for a short period of time without experiencing issues with burning insulation. While not accomplishing the goal of reaching this heat flux, the evaporator plate was able to remove heat loads varying from 400 - 3200 W under various tilt and pitch angles along with pressure restrictions. The resulting vertical flat plate double-sided evaporator is the first of its kind specially configured for cooling electronics. The resulting capillary assisted thermosyphon loop has proven very robust and reliable under a variety of imposed conditions with most of its success attributed to the flat plate evaporator and its internal geometry.

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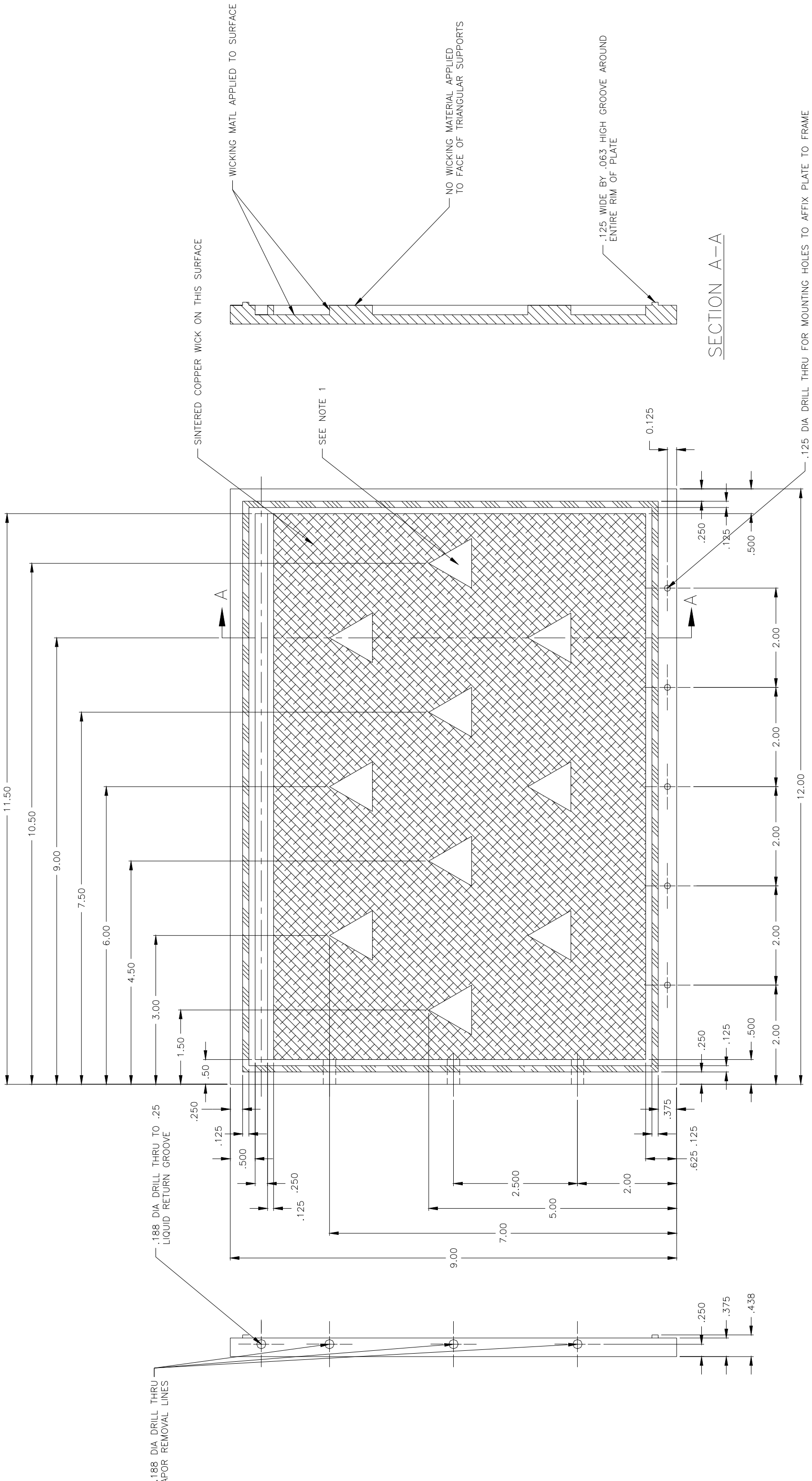
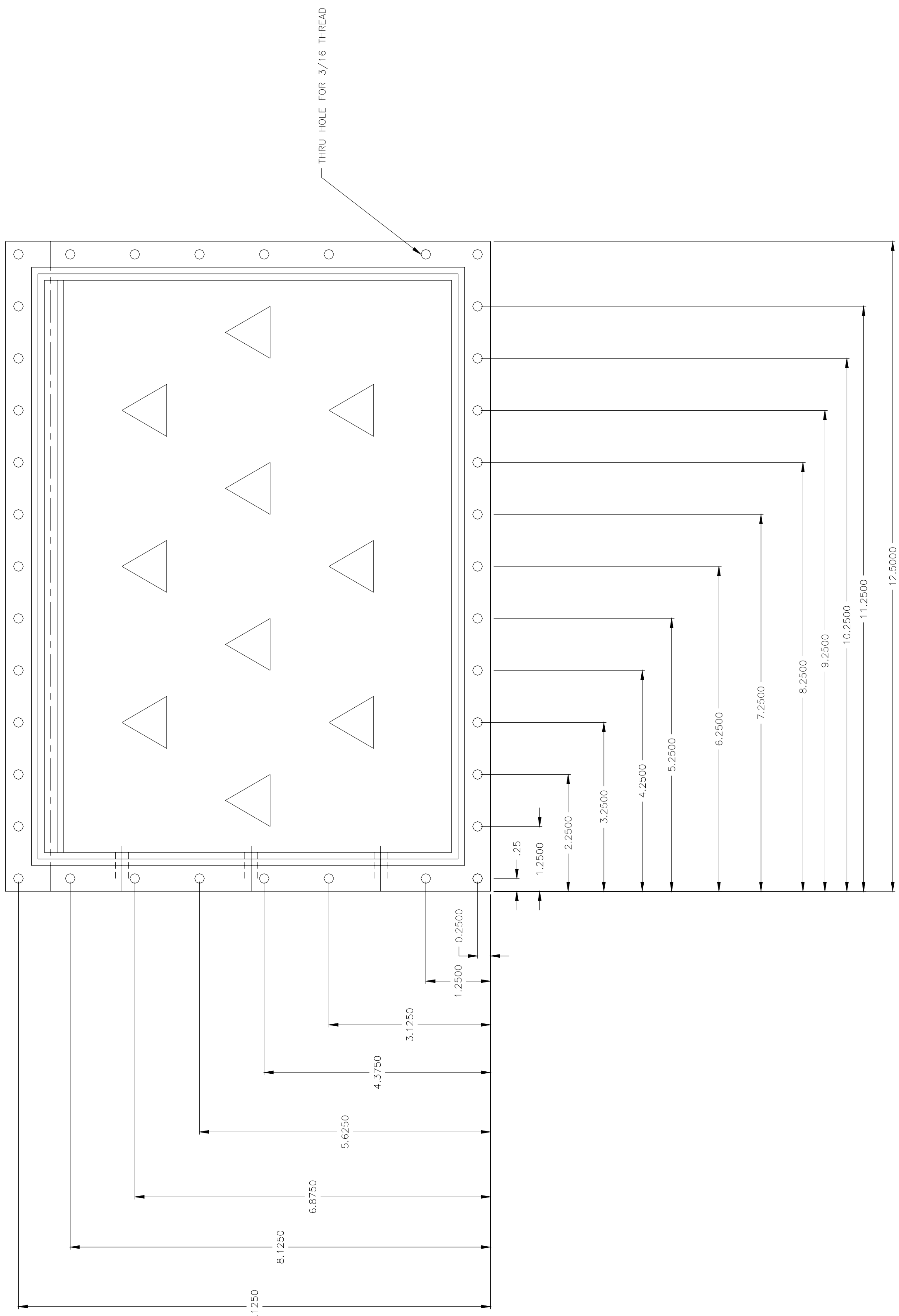
APPENDIX 1: COMPILATION OF CAT LOOP DESIGN DRAWINGS

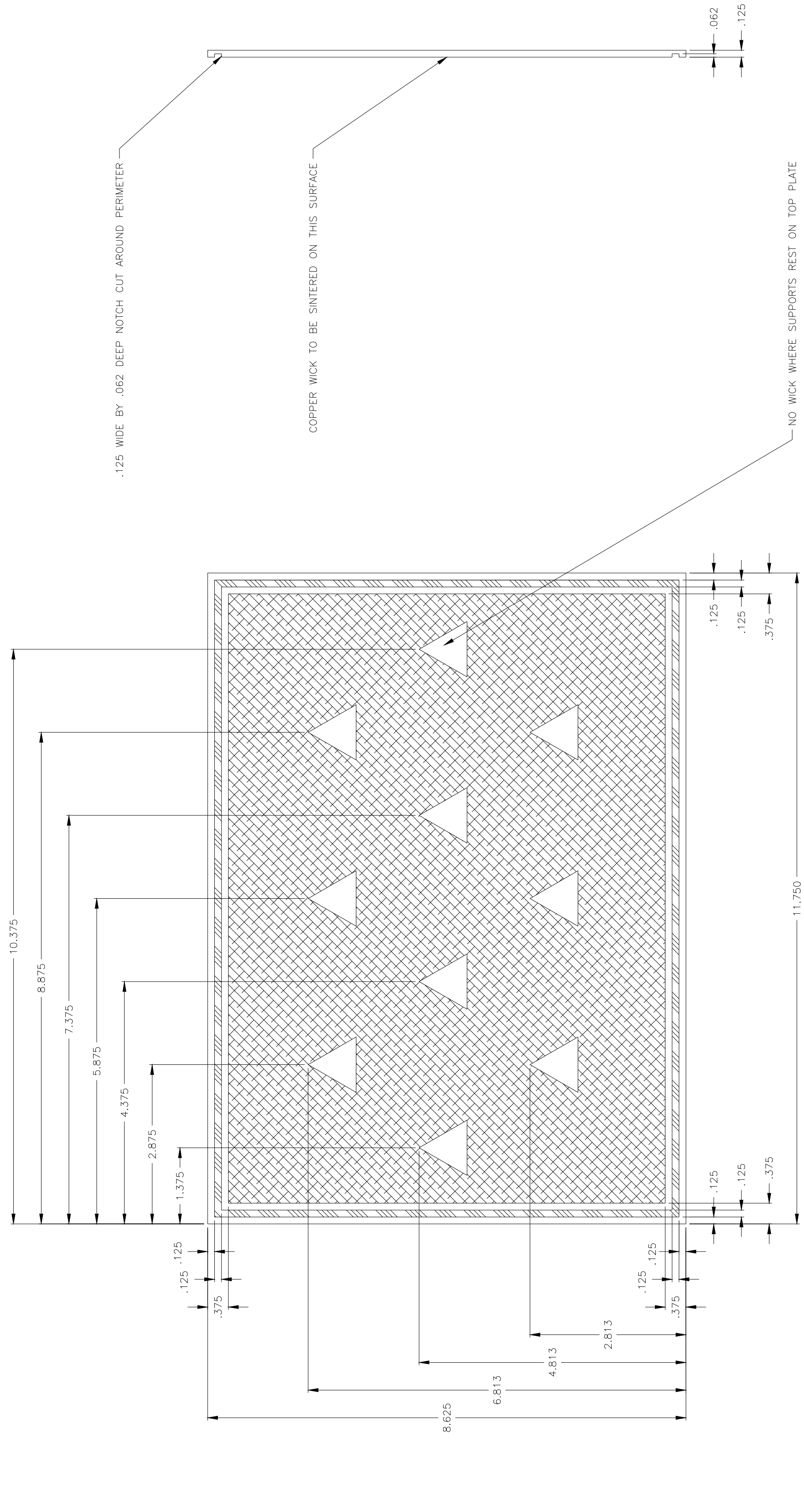
Evaporator Bottom Plate (Original Design)



Bottom Plate w/ Bolt Hole Pattern

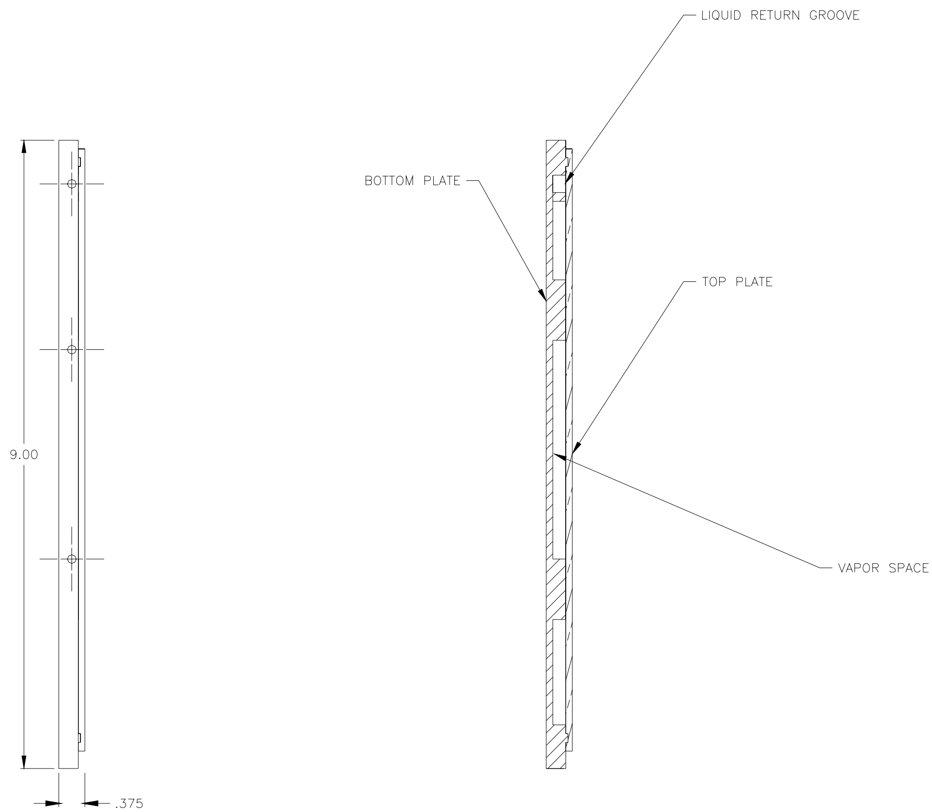


Top Plate

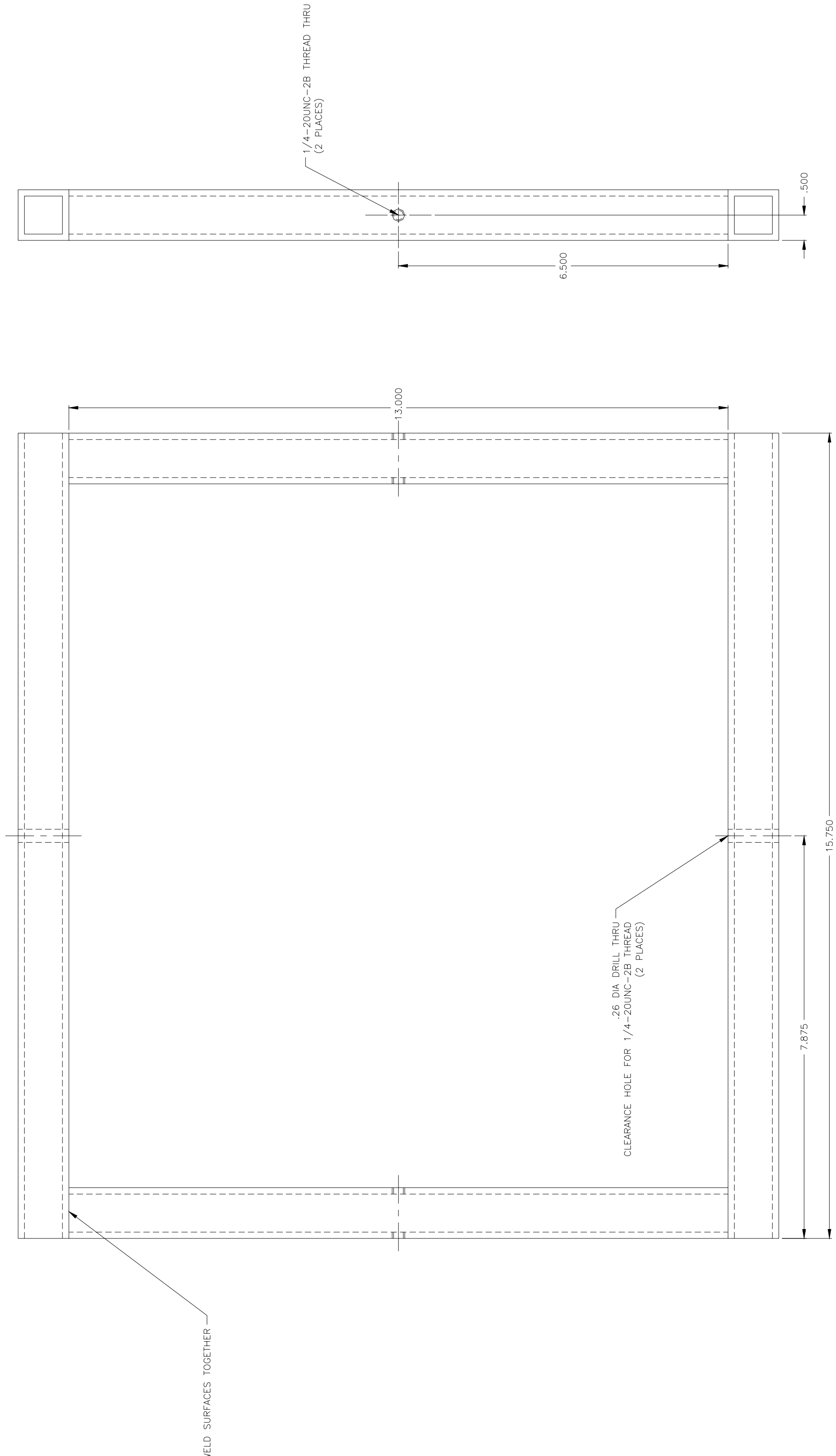


.125 WIDE BY .062 DEEP NOTCH CUT AROUND PERIMETER
COPPER WICK TO BE SINTERED ON THIS SURFACE
NO WICK WHERE SUPPORTS REST ON TOP PLATE
.062
.125
.125
.125
.375
11.750
10.375
8.875
7.375
5.875
4.375
2.875
1.375
.125
.125
.375
2.813
4.813
6.813
8.625

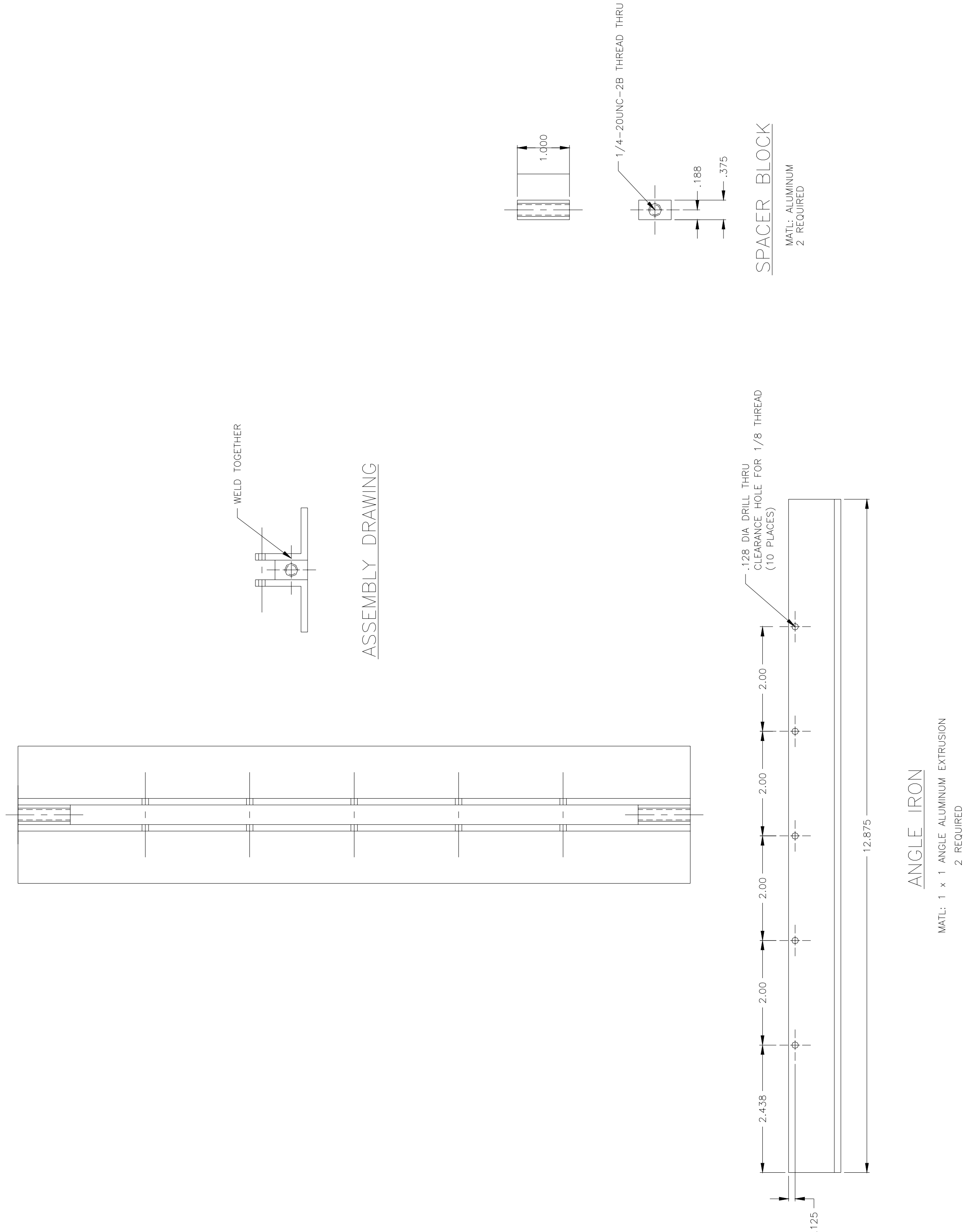
Evaporator Assembly Drawing



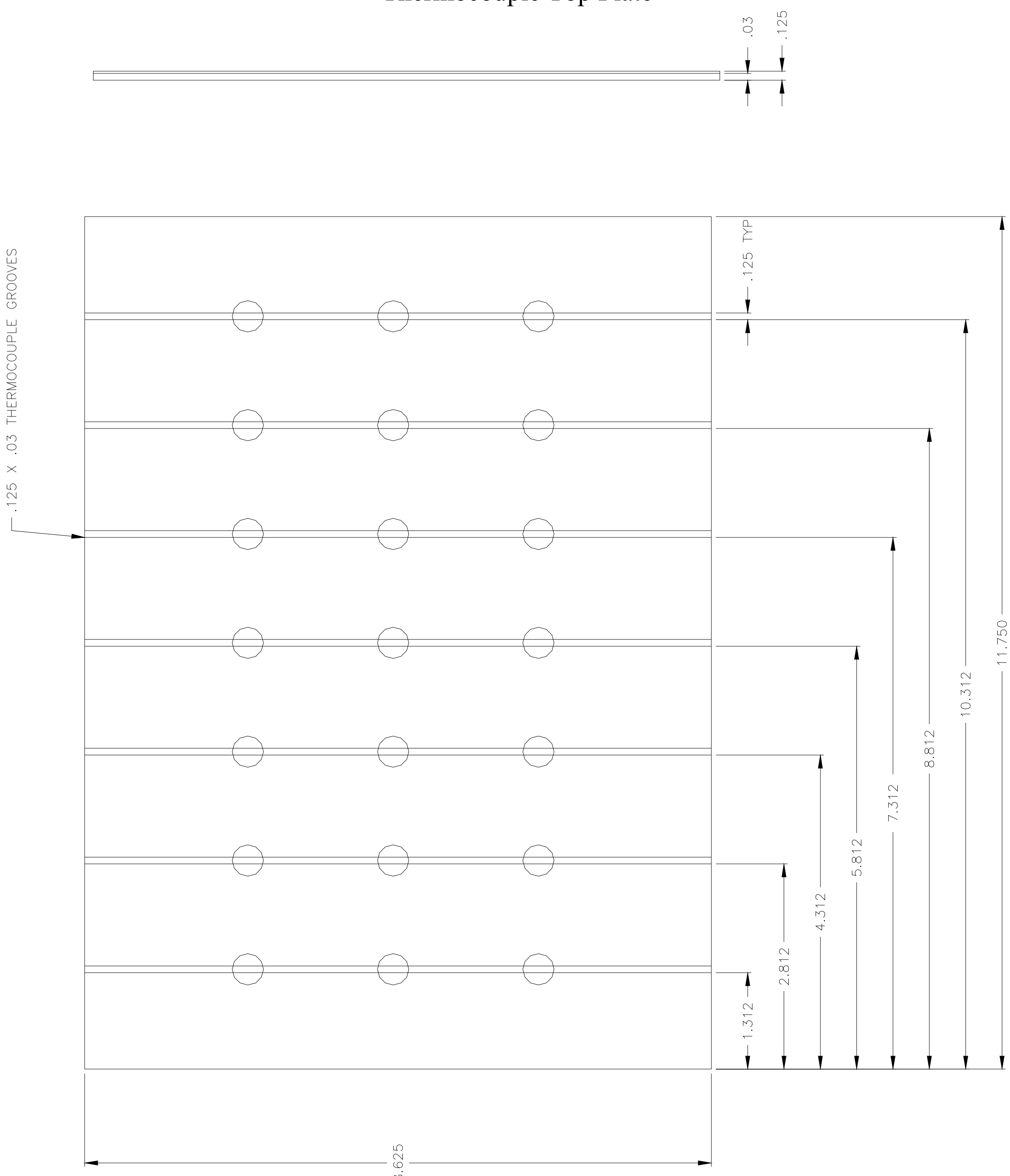
Tilt and Pitch Frame



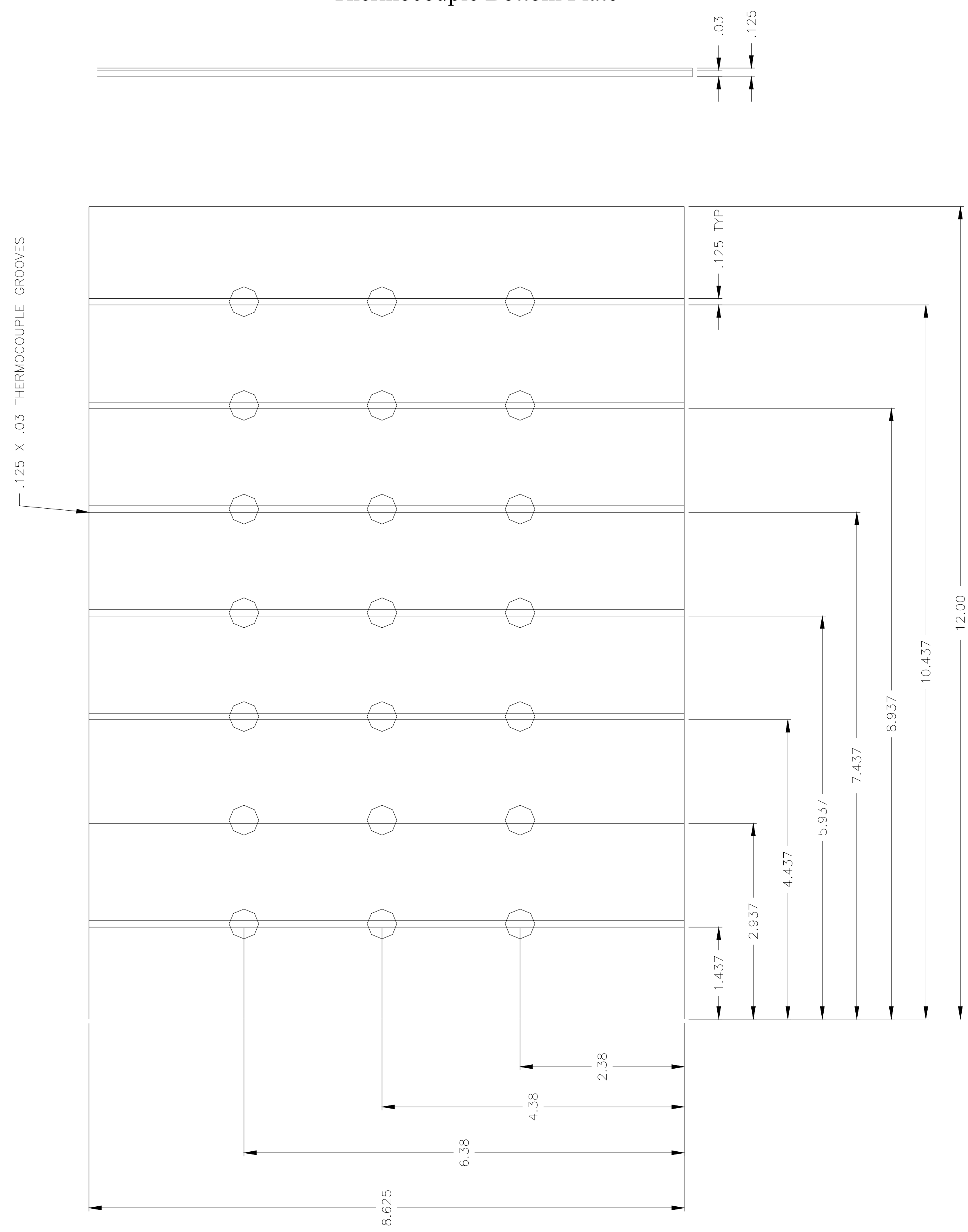
Tilt and Pitch Mounting Bracket



Thermocouple Top Plate

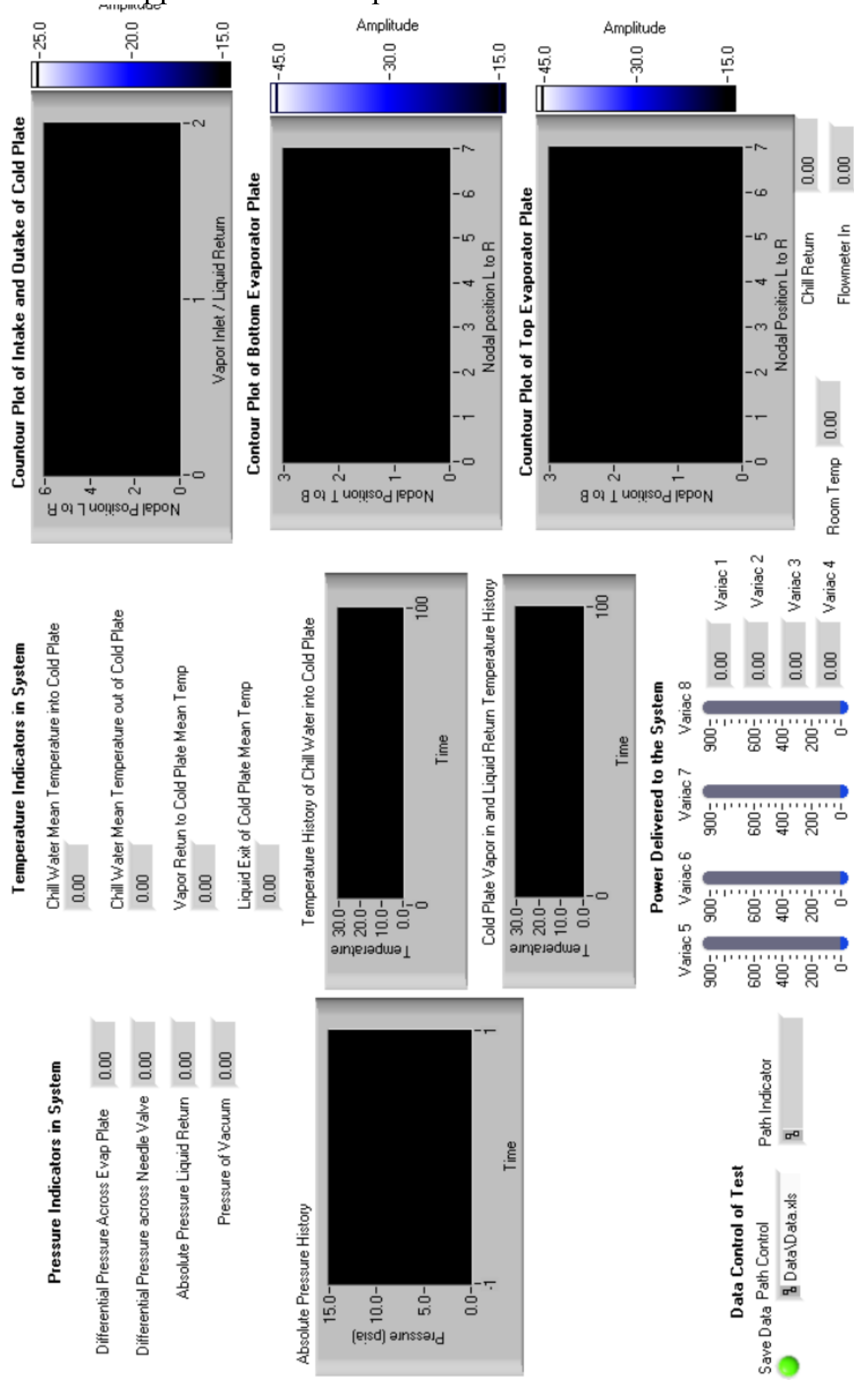


Thermocouple Bottom Plate



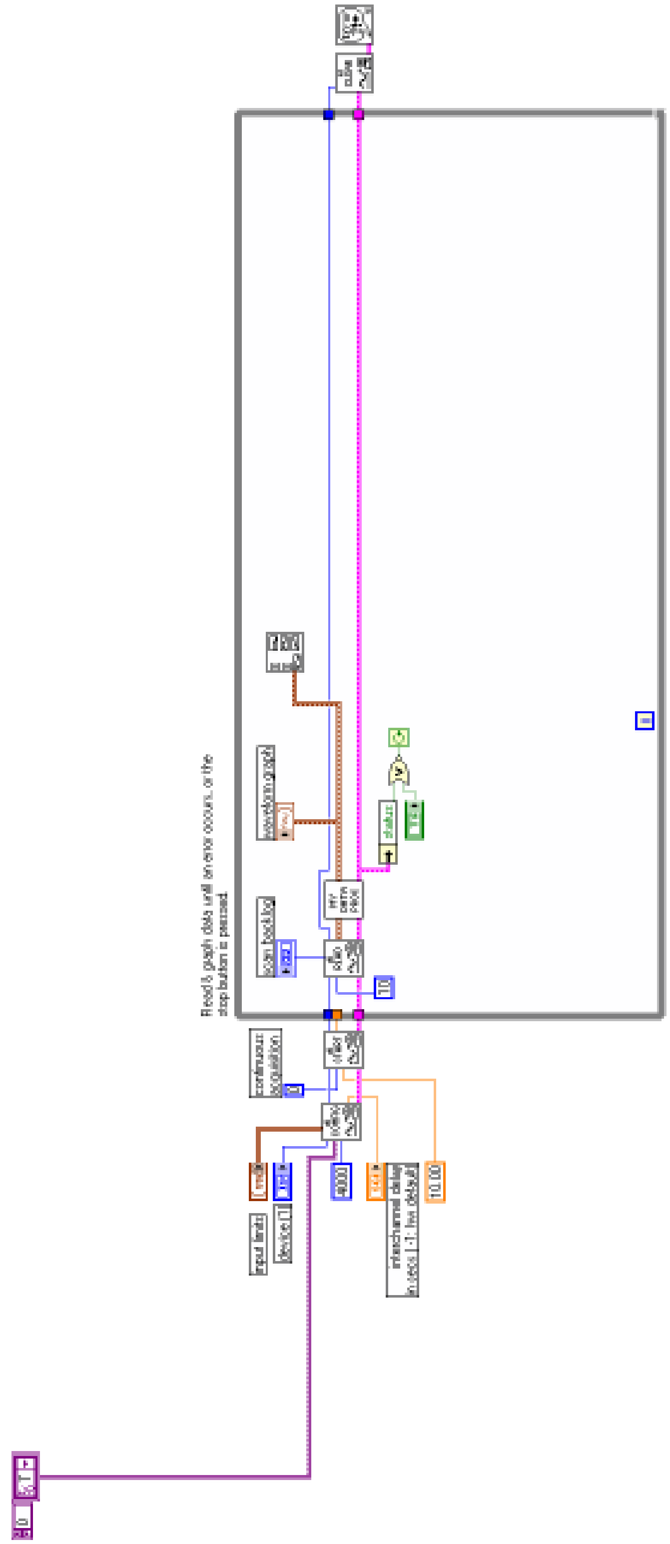
APPENDIX 2: LABVIEW SCHEMATICS

LabVIEW Control Panel and User Interface: The Schematics that follow this page are the wiring and support structure to support this control panel.



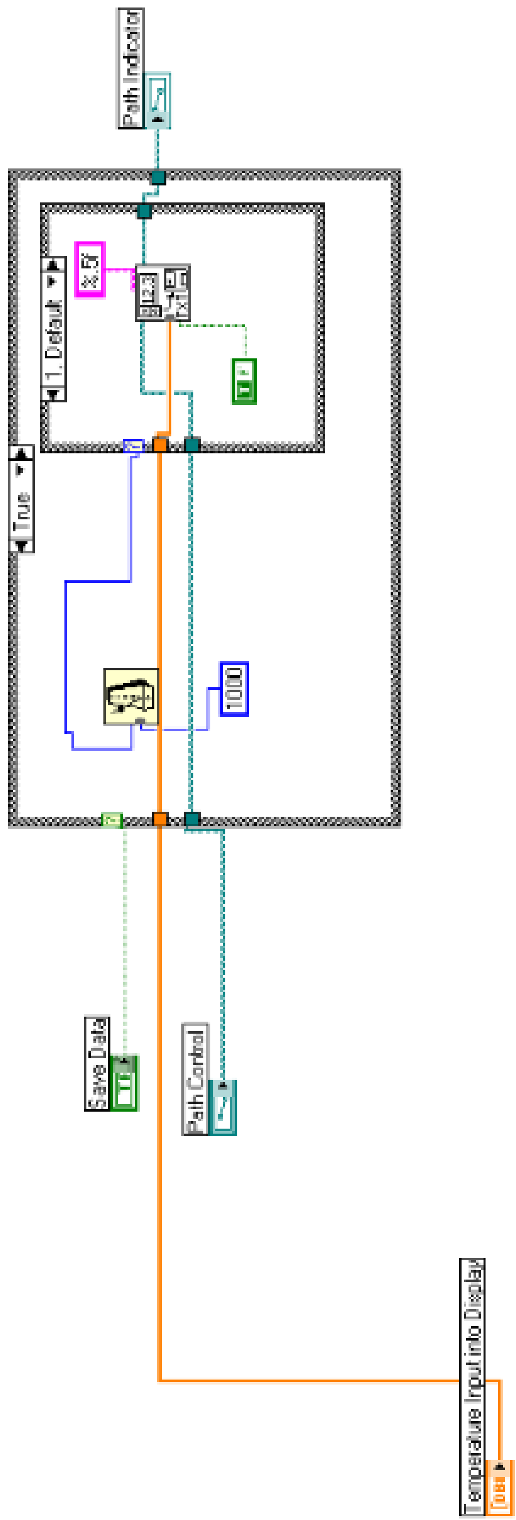
Data Acquisition Elements

Identification, Numbering, and Scan Rate of Input Channels

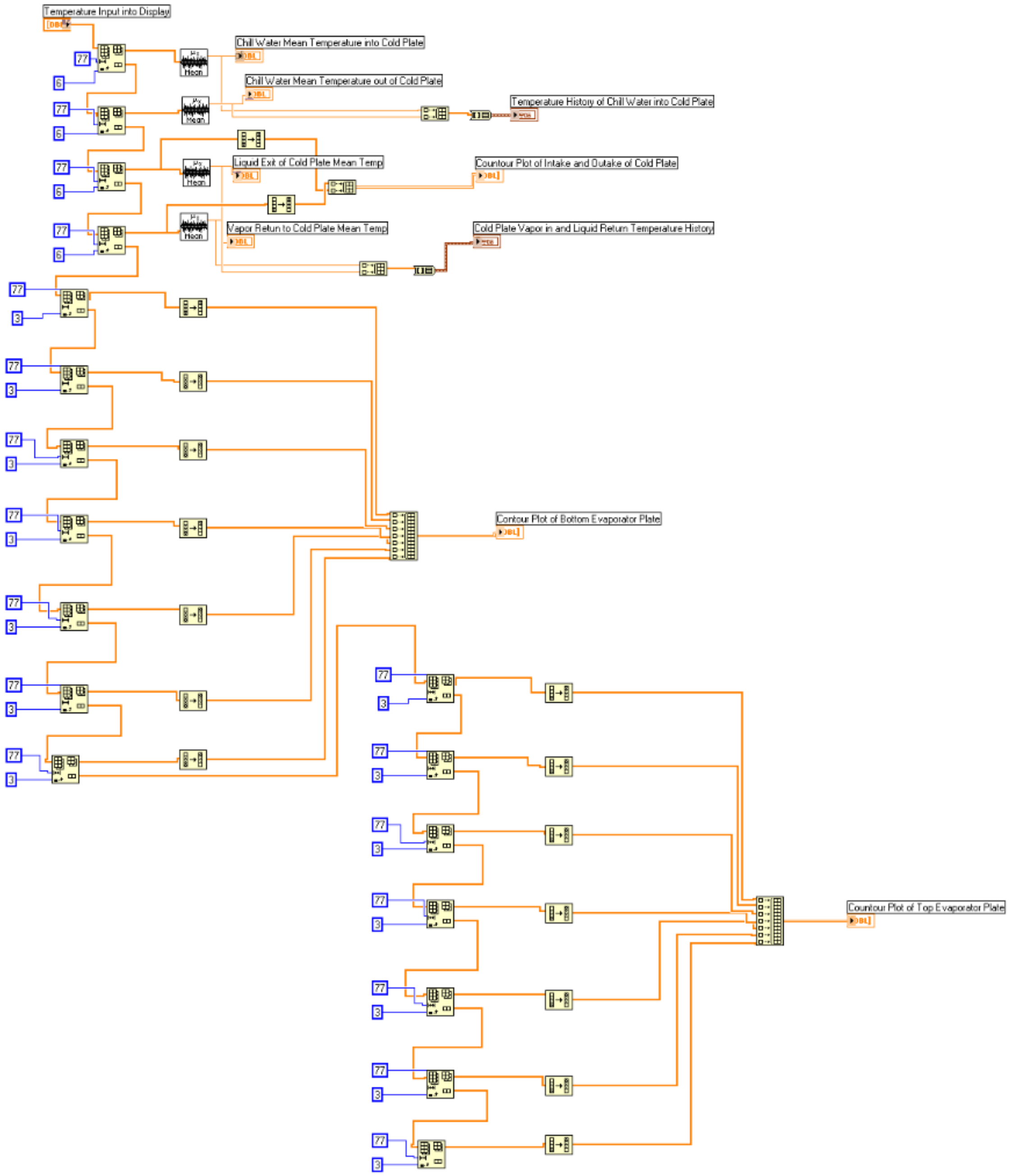


Data Storage

Saving Channel Data into Excel Spreadsheet



Temperature Channels
Configuration of the Contour Plots



Configuration of Pressure and Ammeter Channels